PERFORMANCE STUDY OF MECHANICAL DRAFT COOLING TOWERS

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TO
MY PARENTS

CERTIFICATE

This is to certify that this work on ''Performance Study of Mechanical Draft Cooling Towers'' has been carried out under my supervision and it has not been submitted elsewhere for a degree.

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CYRIL MIRANDA

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NOMENCLATURE

Counter flow:

A	Area of tower cross-section, m^2
a	Interfacial contact surface, m^2/m^3 of tower volume
C_p	Specific heat of air, kcal/kg.$^{\circ}$C
C_w	Unit heat capacity of water, kcal/kg.$^{\circ}$C
d	Designates differential element
G	Air flow rate, kg/hr.m^2 of tower cross-section
h	Enthalpy of main air stream, kcal/kg dry air
h''	Enthalpy of saturated air at water temperature, kcal/kg dry air
h_a	Enthalpy of air-water vapour mixture at the equilibrium wet bulb temperature, kcal/kg dry air
$h_{fg,s}$	Latent heat of vaporisation for water, kcal/kg
K	Overall mass transfer coefficient, kg/hr (m^2 of contact area) (kg water/kg dry air)
K_a	Volumetric mass transfer coefficient, kg/m^3 hr(kg/kg)
L	Water flow rate, kg/hr$\cdot m^2$ of tower cross-section
$\dot{L}$	Total water flow rate, kg/hr
l	Water flow rate, kg/sec$\cdot m^2$
Le	Lewis Number, dimensionless $(\frac{\alpha}{K\ C_p})$
Q	Amount of heat to be dissipated in a cooling tower, kcal/hr
T	Water temperature, $^{\circ}$C

T_1	Water temperature entering the tower, $^{\circ}C$
T_2	Water temperature leaving the tower, $^{\circ}C$
T_{wb}	Wet bulb temperature of moist air, $^{\circ}C$
V	Tower volume, m^3/m^2 of tower cross-section
W	Humidity ratio of moist air
α	Coefficient of heat transfer by convection, $kcal/hr.m^2.{}^{\circ}C$
ε_h	Effectiveness of cooling tower as an energy exchanger
μ	Tan^{-1} (L/G), slope of water-to-air flow rate ratio, deg
η	Fraction of the packing area aV covered by the flowing water

Cross flow:

G	Air flow rate, $kg/hr.m^2$ of vertical air inlet area
H_a	Enthalpy of saturated air, kcal/kg dry air
H_w	Enthalpy of saturated air at water temperature, kcal/kg dry air
i, j	Array notation for point being considered
K_a	Volumetric heat transfer coefficient, $kcal/hr.m^3$ (kcal/kg dry air)
L	Water loading, $kg/hr.m^2$ of horizontal water inlet area
T_w	Water temperature, $^{\circ}C$
$\bar{X}$	Dimensionless co-ordinate in a cross flow cooling tower in the direction of air flow
X	Packing depth, in the direction of air flow, m
$\Delta\bar{X}$	Mesh size, dimensionless
$\bar{Z}$	Dimensionless co-ordinate in a cross flow cooling tower in the direction of water fall

Z	Packing height, m
$\Delta \bar{Z}$	Mesh size, dimensionless

SOME OPERATING TERMS PERTAINING TO COOLING TOWERS

Approach: The difference in degrees centigrade between the temperature of the cold water leaving the cooling tower and the ambient air wet bulb temperature.

Cell: The smallest tower subdivision which can function as an independent unit. Each cell has its own air and water flow system with one or more fans or stacks.

Cooling range: The number of degrees in centigrade, water is cooled in the tower. It is the difference between the inlet hot water temperature and the outlet water temperature.

Drift: The loss of water in the form of fine droplets being carried away by the exhaust air. Drift is measured as the percentage of the circulating water.

Evaporation-loss: The amount of water evaporated from the circulating water into the atmosphere. Expressed as the percentage of total water flow rate.

Fogging: When the warm saturated air is discharged out of the tower, it comes in contact with the colder atmospheric air resulting in the formation of fog.

Heat load: The amount of heat dissipated in a cooling tower in kcal/min from the circulating water.

Make up: The quantity of water added to the basin to replace water lost by evaporation, drift, blow down and leakage (if any).

Recirculation: The entry of a portion of the discharged air along with atmospheric air into the air inlet.

ABSTRACT

A cooling tower is an enclosed device for the evaporative cooling of water by contact with the air. This is achieved partly by an exchange of latent heat resulting from the evaporation of some of the circulating water, and partly by a transfer of sensible heat.

Cooling tower industry has a very competitive market, and hence the refinements in this field have been considered of primary importance. The manufacturer is required to have a set of guaranteed performance curves to refer in selecting a cooling tower for a particular application under specified conditions. These cover the types of tower and packing which are carried by an individual firm. These also cover the operating conditions such as water flow, cooling range, cold water temperature, wet bulb temperature etc., in order to design a cooling tower.

The cooling tower manufacturers in our country do not have guaranteed performance curves and have to, therefore, either guess the tower size or they depend on their foreign collaborators. The present work has been undertaken as an attempt towards solving this problem of the cooling tower industry. Performance curves have been drawn for both the counter flow and cross flow mechanical draft cooling towers for power plants, fertilizer and air conditioning plants, designed to be located in big industrial cities of India. Generalised computer programs have been developed,

based upon the cooling tower theories already developed. Results have been analysed and discussed.

It is expected that the performance curves developed in the present work should be of great help to the cooling tower industry in the country and to the buyers in selecting and predicting tower performance at varying operating conditions.

CHAPTER I

INTRODUCTION

Conservation and reuse of processed water have become a necessity world over. Large industrial water users are power generation plants, chemical plants, steel plants, petroleum refinaries, atomic power plants, air-conditioning and refrigeration industry etc. An immense quantity of water is used by most of the above industries for cooling purposes, for example, in condensing the exhaust steam in power plants, in liquefying the chemical products in vapour state and, in many cases, in preventing overheating of the machinery parts which are exposed to high temperatures. The standards required of the cooling water as regards its temperature and quality, i.e. its contents of impurities, may vary considerably depending on the purpose of the cooling water.

It is necessary that the temperature of cooling water should not exceed a certain prescribed value for a particular process plant and that its content of impurities should not result in the formation of deposits in the system or in corroding the metal parts. These requirements are dictated by the nature of the production processes and by the need for reliable and economical operation of the plant concerned. A rise in temperature of the cooling water used in a steam plant, for example, increases the fuel consumption in power production and lowers the plant's capacity; in

refrigeration, it effects the coefficient of performance of the plant and in oil refinaries and chemical plants, it lowers the yield of the products. Similar effects are experienced because of the impurities present in the cooling water. Another of the more important conditions on the temperature of the cooling water is that it should not be heated to a very high temperature in the condensing plant i.e. large quantities of heat should be transferred at a low temperature to achieve the most efficient results. This necessitates a very high consumption and continuous supply of fresh water.

Due to the increase in power demand, expansion of industries, etc., the use of water has more than doubled in the past decade, and its resources, everywhere, are limited. One has to mainly depend upon seas, rivers, lakes, ponds and underground systems as sources of water supply. In tropical countries like India even these resources are not easily available in most of the parts. Underground reserves [1] are becoming exhausted because of heavy exploitation over a considerable period and a lack of legal control. Thus, at many places, rain water is the only source left.

When a sufficiently large supply of water is available from the sea, river, lake etc., a continuous water supply system is mostly used, in which the water taken from the source is used once for the cooling purposes and is then discarded. Considerable variations

in the water level in a river or lake are found and sometimes the water is to be transported over large distances or to a great height. Extensive mineralization of water or contamination by chemically - aggressive impurities, requires an expensive continuous purification process. Under such circumstances, when the procurement of usable water from its source is so expensive, cooling by a continuous fresh water supply cannot be adopted.

Problem is not only to obtain a continuous fresh supply of water in a sufficient quantity from the source for cooling purposes in industries, but an equally important problem is to find a way of discarding water that has been used for cooling. Temperature is a primary factor [2] in the solubility of atmospheric oxygen in water, and in all chemical and biological processes occuring in water. The discharge of large quantities of heat to receiving waters may affect important processes adversely and result in a less desirable quality of water. Of all water uses, the propagation of aquatic life is perhaps the most affected by temperature. For this reason, establishing temperature standards to protect fish and their food organisms has been of primary importance.

Reuse of industrial cooling water is a big step not only in the water conservation program, but it also eliminates the problems of discarding hot water resulting in thermal pollution and is very economical too.

One of the methods to cool water is to use air as an external heat absorbing medium when hot water is being circulated through the tubes. Water acts as an intermediate heat - carrier between the plant being cooled and the outside air. The heat removal takes place by conduction through the tube wall and by convection from the outer surface of the tube to the moving air.

The use of air as an external heat - absorbing medium has not been adopted very widely. The average value of the convective heat transfer coefficient of air varies from 2.7142×10^{-3} kcal/sec - m^2 - ^oK (2 Btu/hr - ft^2 - ^oF) to 13.571×10^{-3} kcal/sec - m^2 - ^oK (10 Btu/hr - ft^2 - ^oF) depending upon the velocity and temperature of air [1]. Hence, due to the poor heat transfer from the surface being cooled to the air, the cooling surface required is many times greater than that required for cooling by water. Also, the specific heat of air is low and, hence, a high power consumption is required for fans to supply large quantities of air. Nor it is possible with this device to cool water below the ambient temperature of air. Therefore, this device cannot be used as an inexpensive and effective device for cooling water in industrial units.

Considerable increase in the rate of heat transfer between the atmospheric air and the circulating water can be achieved by bringing hot water into direct contact with moving air. This employs the principle of evaporative cooling of the water. In evaporative

cooling, a part of the water gets evaporated and the water vapour thus produced carries away with it heat which is called the latent heat of vaporization. The evaporative cooling approach is basically a water saving technique.

The thermodynamic principle of evaporative cooling is that water must have heat to change from the liquid to the vapour state and when water evaporates, this heat is removed from the water remaining in the liquid state, dropping its temperature. It takes approximately 560 kcal (1000 Btu) to evaporate 1 kg (1 lb) of water. Therefore, the available cooling effect for an evaporative type cooler is about 560 kcal per kg of water evaporated.

In a once - through system using city, well, or surface water, each kg of water circulated will pick up only 1 kcal for each $1^{\circ}C$ of temperature rise, since only ''sensible'' heat gain through the unit being cooled is involved. If the temperature requirements of the heat source allow for a $10^{\circ}C$ temperature rise, the heat pickup would be 10 kcal/kg of water circulated. Therefore, in an evaporative cooling device 1 kg of water does the same work as 56 kg of water in a once-through system. Theoretically, evaporative cooling requires only about 2 per cent of the water compared to a once-through system. The removal by air of latent heat plus sensible heat, makes the water cool by evaporation in a cooling

device, as shown in figure (1.1).

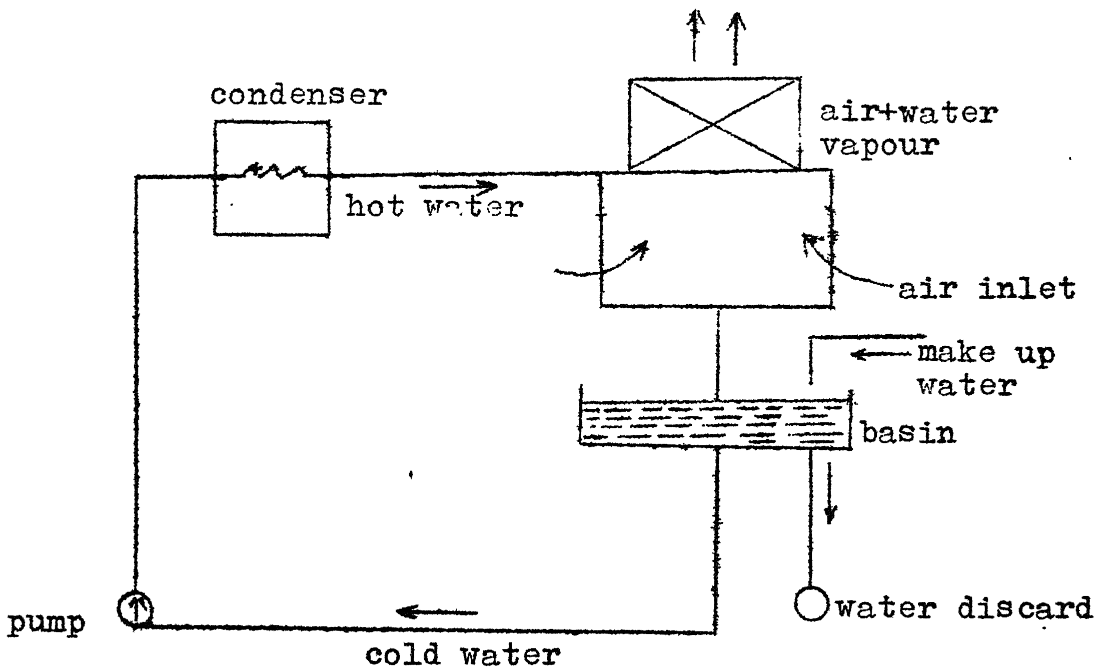


Fig.(1.1), Schematic diagram of circulating cooling water-supply system.

The air is chosen as a medium for cooling water because it is so freely available in unlimited quantities. It not only absorbs the water waste heat load but also carries it away and dissipates it into the atmosphere without altering the atmospheric conditions. When evaporative cooling is used, the cost of water collection from the source and its purification is incurred only once and the waste water problem is almost fully eliminated.

Because of these advantages, evaporative cooling of circulating water has predominated in recirculation

water - supply systems.

Figure (1.2) shows a portion of a psychrometric chart and indicates the psychrometric analysis of the air path through the evaporative cooler. The true path is approximated by the curved dotted line from point A (entering air conditions) to the point C (leaving air conditions). This actual path will vary either above or below that shown, depending on the unit design and the ratio of air to water flow.

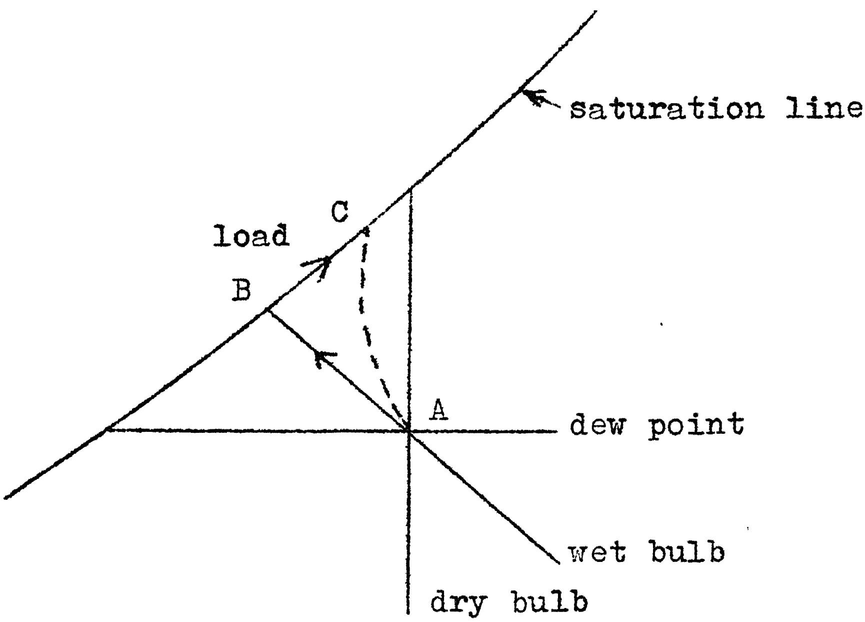


Fig. (1.2), Cooling effect of air passing through evaporative cooler.

For purposes of explanation, the air path is broken down into two vectors, lines AB and BC. Air

enters the evaporative unit in an unsaturated ambient condition (point A). Before reaching the heat transfer surface, it is saturated without change of total heat content as it travels to the point B. Passing across the heat transfer surface, the air absorbs heat from falling water. As some of the water evaporates - the heat content of the air is increased as the temperature of the falling water is reduced. Since the air is continually being washed by the water, the process follows the saturation line to the final leaving air temperature (point C). Travel from point A to point B is adiabatic - there is no cooling of the water. There is only a conversion of air sensible heat to latent heat as the air temperature drops to that of the wet bulb temperature. The effective heat removal takes place between the points B and C along the saturation line. Thus, the wet bulb condition of the entering air at the point A is the only factor affecting the cooler performance.

The principle of evaporative cooling was well known since ancient times in India and other Asian countries. Only recently, however, it has been put to practical industrial application in these countries. In U.S.S.R., U.S.A., and European countries where industrial development started much earlier than in Asian and African countries, evaporative cooling gained a great deal of importance and has been long analysed and studied scientifically.

Among the earliest theoretical and experimental

studies made in England on this subject, I.V. Robinson's work in 1907 [3] on the theory of cooling towers appears to be the first in the field. Among the earliest studies made in the U.S.S.R., were those of A.N. Arefóv in 1925 at the F. Dzerzhinskii All - Union Power Engineering Institute on cooling towers [4] and those of N.M. Bernalskii in 1929 on cooling ponds [4]. These studies formed the basic of the development of the theory and methods for the thermal design of coolers employing evaporative cooling of water. It was only in 1926, that a successful theoretical analysis of heat and mass transfer in cooling towers was made by Merkel in Germany [3]. The Carrier charts for calculation of moist air relationships were developed in U.S.A. in 1911 and a similar chart by Mollier in Europe in 1923. Extensive efforts to explore the field of industrial cooling plants were not made until after the incidence of second world war. Design of various types of cooling devices and their use for industries, homes and buildings become familiar all over the world around the 1930's.

The main types of water cooling devices are classified as follows:

1) Ponds
   (a) Cooling
   (b) Spray

2) Closed Coolers (natural draft)
   (a) Spray - filled
   (b) Splash - filled and

3) Tower Coolers
   (a) Natural draft chimney tower
   (b) Mechanical draft tower
      (i) Forced draft
      (ii) Induced draft (counter-flow, cross-flow)

1(a) Cooling Ponds: It is the cheapest and the simplest method of cooling water. It consists of a comparatively large pond in which cooling takes place by air contact at the surface. In order to achieve the best results, the total area of the water surface should be as large as possible. This is achieved by having the inlet and outlet points placed as far apart as possible and by raising the water level in the pond.

The equation for estimating the size of an evaporation pond needed for a specific incoming flow rate is [5]:

$$D = \left(\frac{525,600\ \dot{L}}{7.48 \times 43,560 \times A} - \frac{E_n}{12}\ T \right) \qquad (1.1)$$

where
D = Depth of pond, ft.
$\dot{L}$ = Flow of processed water into the pond, gpm.
A = Area of pond, acres.
E_n = Net evaporation (evaporation - rain fall), inches/year.
T = Time, years.

conversion factors:
525,600 = mins per year
7.48 = gal/ft^3

$43,560 = ft^2/acre$
$12 = in/ft$

A cooling pond has the following advantages:

(i) Cooling pond may be constructed at a very low cost by pushing up a certain height of the earth.

(ii) It may operate for a long period without requiring any make-up water and with low maintenance cost.

However, the use of the cooling ponds is quite limited because of the following disadvantages:

(i) The heat transfer rate from a cooling pond is very low. The heat dissipated from a still pond averages 17 kcal per hour per square meter of water surface of a still pond, per degree centigrade temperature difference between the contacting air and water surface.

(ii) Large areas needed for the ponds create serious problems particularly in big industrial cities.

<u>1(b) Spray Pond</u> : A basic modification of the cooling pond concept involves the use of spray ponds. The spray pond cooling system pumps heated water through spray nozzles, which divide the water into small droplets, thus increasing the effective area for evaporative heat transfer to the atmosphere. Cooling occurs in the spray pond as the water is propelled upward and

falls to the surface of the pond. The pond acts largely as a collecting basin.

The amount of cooling to be gained in direct air-water contact of the spray system ultimately depends on the air-temperature, humidity and wind conditions. Higher wind speeds effect more efficient heat transfer to the atmosphere. A louvred fence may be provided to reduce the loss of water with the outgoing air.

Although spray ponds are more compact in design and better in their performance than cooling ponds, they still require larger areas and have limited performance capacity because the time of contact of sprayed water and air is too small. Their use, therefore, has nearly completely stopped since the cooling towers came into existence in the 1920's.

2(a) Spray-filled Cooler (Fig 1.3):

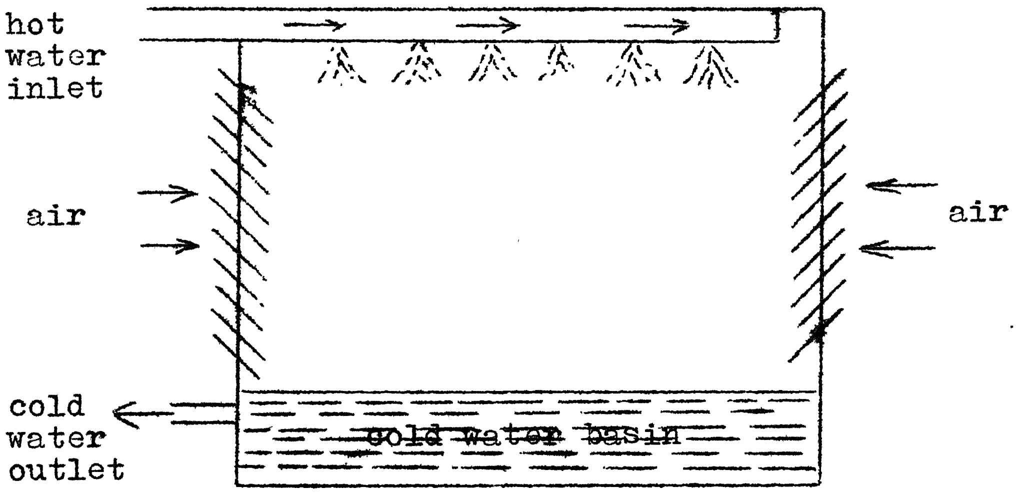


Fig. (1.3), Spray-filled cooler.

This is essentially a narrow small-sized spray tank with elevated nozzles and a high louvre fence. The air movement through the louvred side is fully dependent upon wind conditions. Since wind normally blows in a horizontal direction, flow of air is crosswise to the flow of water.

The spray nozzles used are of the tangential type which are less liable to become clogged up. The operating pressure before the nozzles is from 3 to 5 meters of water, or sometimes, between 12-14 meters of water. Nozzles pointing downwards are used which results in more uniform distribution of the water and reduces drift.

Spray-filled coolers are suited for refrigeration and air-conditioning system applications. Table (1.1) gives the typical dimensions of spray-filled coolers, with nozzles pointing downwards [4].

These towers can be installed in the open where there is no obstruction to prevailing winds. It can be installed on the roof of a building or on a special platform raised for the purpose, if there is a shortage of space.

A spray-filled tower has the following disadvantages:

(i) The approach to the wet bulb temperature

Table (1.1) Typical dimensions of spray coolers.

Nominal output m^3/hr	Area m^2	Dimensions, mm		
		width	length	Height
1.1	0.85	920	920	1830
2.3	1.50	1220	1220	1830
4.5	1.90	1220	1530	2750
10.0	3.40	1830	1830	2750
20.0	4.60	2140	2140	3660
30.0	8.60	2140	4020	3050
50.0	20.60	2140	9660	2750
50.0	21.40	2750	7780	2750
50.0	21.00	4020	5240	2750
100.0	36.80	2750	13400	3660
100.0	38.80	4020	9660	3660
100.0	46.30	4020	11540	3660
150.0	61.50	4020	15300	3660
200.0	76.40	4020	19000	3660
250.0	91.70	4020	22800	3660
300.0	106.80	4020	26600	3660
340.0	122.50	4020	30500	3660

will always be equal to or greater than the cooling range, except when relatively high hot water temperatures are encountered.

(ii) There are windage losses.

<u>2(b) Splash - filled Coolers</u>: The main difference between the splash - filled coolers and spray - filled coolers, is that in the splash - filled coolers, filling is used to increase water break-up and to provide additional water surface to air flow. The cooling efficiency is greater in splash - filled coolers. It's increased cost and maintenance makes it obsolete.

<u>3(a) Natural - draft Towers</u>: Natural-draft towers were the first large cooling apparatus built. It consists of an empty shell made of steel - reinforced concrete structure mostly in hyperbolic designs. These are built very high up to 120 meters with the base diameter 80 meters. At the lower end is the packing through which water trickles, drops or flows in a predetermined manner so as to give up a portion of its heat to the air stream flowing past.

In the hyperbolic-type tower, shown in figure (1.4) the packing occupies the whole of the space between the hot water distribution system and the air inlet position. There is no air distribution system, and the air enters the packing at the periphery of the bottom of the tower.

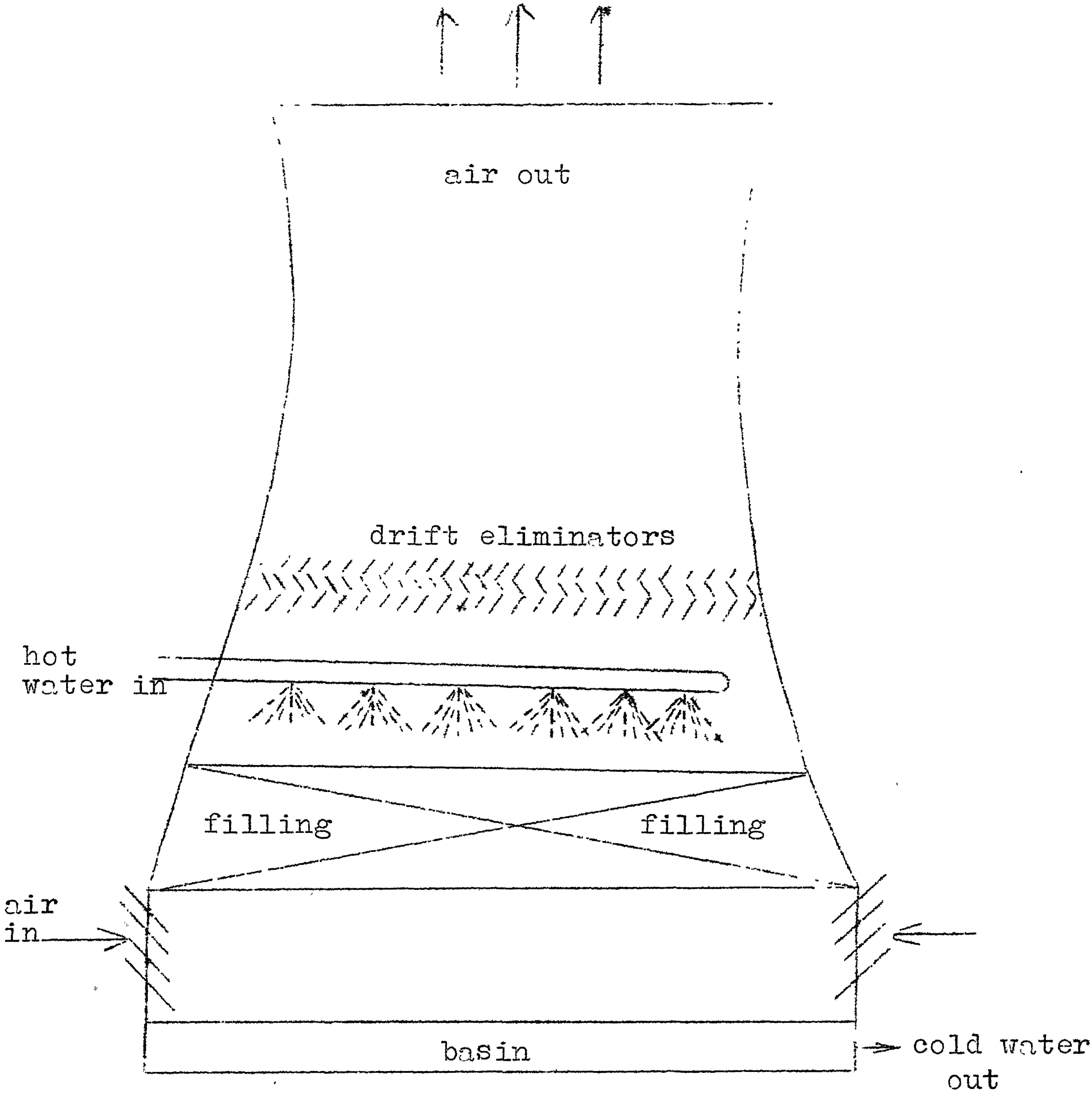


Fig. (1.4), Hyperbolic Tower.

The hyperbolic shape of the tower does not have

any thermodynamic significance. It could be of a cylindrical shape as well.

From an aerodynamic point of view, hyperboloids or cones are better than cylinders [6]. Hyperbolic tower is still better than the conical as it directs the entering air more smoothly towards the centre, and the upper rim tends to produce a stronger upward draft than the inclined straight rim of the conical. The hyperbolic tower, since it is a doubly curved shell has a great strength and shearing stresses are eliminated.

Air flow in the hyperbolic tower is produced due to so many factors. The difference in the specific weights of the cold dry air outside and the air inside the chimney, and the temperature and humidity raised because of its contact with warm water are responsible for the air-flow movement in the hyperbolic tower. The most significant contribution is due to the draft created by the height of the chimney. The greater the height, the greater is the draft achieved. The average velocity of air above the packing is of the order of 1.5 - 2 meters/second.

The main advantages of natural draft cooling towers are the following:

(i) No fans are required to blow air in the tower. This not only eliminates the capital investment for the mechanical

equipment and the related electric control, in comparison with mechanical draft towers, but it also greatly reduces the expenditure towards the operation and maintenance costs.

(ii) They can practically never break down.

(iii) They can cope with tremendous water loads.

(iv) Ground fogging and recirculation of warm air are practically avoided because of the large height of the tower.

(v) Loss of water due to drift is negligible.

The principal disadvantages are:

(i) The great height necessary to produce the draft.

(ii) Inlet hot water temperature must be kept hotter than the air dry bulb temperature.

(iii) Exact control of outlet cold water temperature is difficult to achieve.

<u>3(b) Mechanical - draft Towers</u>: Mechanical-draft cooling towers have a relatively simple construction, although the heat transfer processes which take place within them are extremely complex. Except for ambient air wet bulb temperature, they are virtually independent of atmospheric conditions. They can cool water to temperatures below ambient with low capital and running costs in a comparatively small space.

There are various types of mechanical - draft

cooling towers. Figure (1.5) illustrates the main components common to most towers.

(i) Casing : This is a structure which encloses the heat transfer process and provides a support for the other main items.

(ii) Fan : In mechanical-draft towers, a fan is fitted to move the required amount of air through the water to be cooled.

(iii) Drift Eliminators : These are placed at or near the air outlet, and prevent droplets of water from being carried out of the tower by the air stream.

(iv) Water Distribution System : For maximum effect the water entering the tower must be spread evenly over the top of the packing. Nozzles are used to atomise the water, for a spray distribution system and trough or weir, where the water spreads by gravity.

(v) Packing : This provides a large water surface area to assist heat transfer. This may be either splash packing or film packing. The depth of the packing may be as much as 7 to 8 meters. The two types of film packing, now in general use are:

(a) grid packing
(b) plate packing. This was developed to increase still further the heat transfer surface per unit volume. It consists of

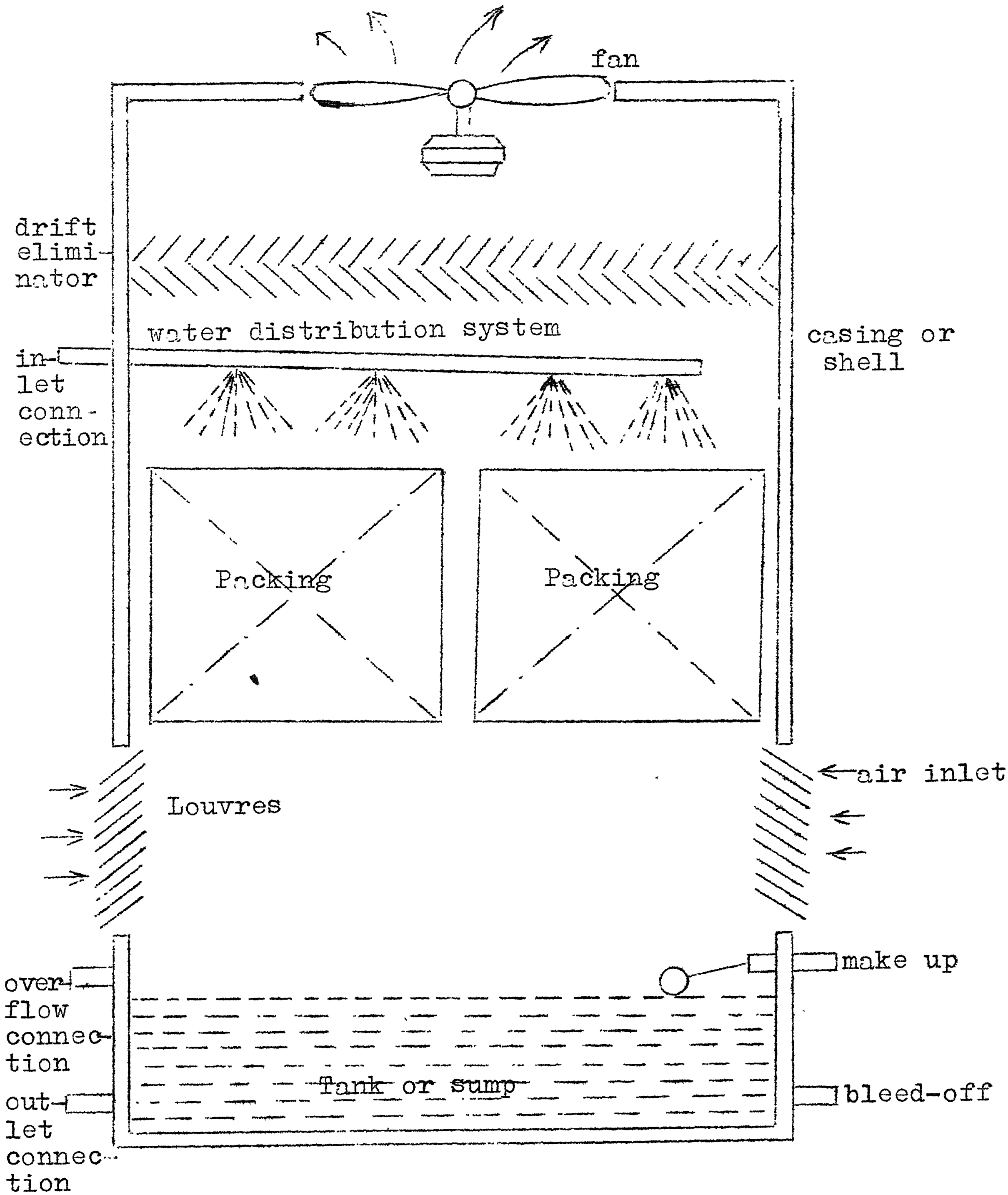


Figure (1.5), Main components of a mechanical draft cooling tower.

closely spaced vertical plates, each plate being corrugated in some manner.

Most splash and grid packings are of timber (specially - treated redwood), although they are also made with T-shaped plastic sections.

Film plate packings have been made in galvanized mild steel, anodised aluminium, stainless steel, cement, asbestos and various plastics [7].

The mechanical-draft cooling tower uses fan power to push the outside air into the tower if it is a forced draft or to exhaust warm air outside, if it is an induced draft.

Forced draft : In forced draft towers, the fans and the electric motors are easily accessible for maintenance. Vibration is kept down because the mechanical equipment is near the ground and on a solid foundation. Some of the disadvantages are, because of the low height of these towers. The chances of the hot, humid exhaust vapours coming out of the top of the tower, to be recirculated are nearly cent percent. The size of the fan is also limited to about 4 meters or less to suit the tower design, which implies that installation of a larger number of towers can meet the demand.

These disadvantages have been overcome in induced

draft towers and, hence, induced draft towers have gained preference and special attention by designers, sellers and users in the recent past.

Induced draft (Figure 1.5) : This has the fan located at the top of the tower to push out the warm air of the tower.

This has been further sub-divided as:

(i) Counter-flow.
(ii) Cross-flow.

In counter-flow type, air is drawn up vertically against the water falling down in the opposite direction through the packing. Maximum performance is achieved in this tower because the coldest water comes into contact with driest air and the warmest water contacts the most humid air, thereby, having the maximum enthalpy gradient at the top at all the sections.

In the cross-flow type, air enters the packing horizontally throughout its height through the louvres while water is falling down across the packing.

Both the types of towers have advantages and shortcomings.

From a thermodynamic standpoint, the counter-flow type is more efficient than the cross-flow type [8] because its enthalpy potential difference is higher.

Lower static pressure is encountered in the cross-flow tower because not all the air passes through the entire fill section reducing operating and maintenance costs. The cost of both the types is about the same.

The cross-flow arrangement has an advantage over the counter-flow system that the drift losses are less; resulting in lower horse power requirements.

The cross-flow design has a larger plenum area [9], and is more susceptible to biological attack. Its open-pan distribution system allows the development of slime and algae, which must be controlled with chemicals, but cleaning the nozzles in a cross-flow tower is simple. The cross-flow tower can be easily altered to increase water flow.

The counter-flow tower has a closed-pipe distribution system to control algae, but is harder to clean.

The counter-flow tower performs better than the cross-flow type with fans off because of the natural draft produced by fewer air intakes. The natural draft is particularly important in the winter or in the evening when more desirable psychrometric conditions exist for the tower.

In India, the first cooling tower was manufactured for the Ahmedabad Electricity Company by Gammon India Ltd., Bombay around thirties. Prefabricated cooling towers for the plants like chemical industry

and for central air conditioning plants continued to be imported in the country till recently. It was only in the last decade, that the following concerns [1] started manufacturing over 400 big and small complete range of cooling towers per annum in the country with different foreign collaborations.

Table (1.2) shows the main cooling tower manufacturers in India and their collaborators.

Table (1.2) Main cooling tower manufacturers in India and their collaborators.

Sl.No.	Manufacturer	Collaborator
1	Air Conditioning Corporation Ltd., 17 Taratalla Road, Calcutta - 53.	Heenan Proude, U.K.
2	Gammon India Ltd., Prabhadevi, Cadell Road, Bombay - 28.	L.G. Mouchell and Partners, U.K. (till recently)
3	Larsen and Toubro Ltd., Dougall Road, Ballard Estate, Bombay - 1.	Film Cooling Towers Ltd., Brentford, Middlesex, U.K.
4	Paharpur Cooling Towers Pvt. Ltd., 1-B Judges Court Road, Calcutta - 27.	The Marley Company, Kansas City, U.S.A. (till recent past)

Table (1.3) shows some of the important cooling tower installations in the country.

Table (1.3) Important cooling tower installations in India

Sl.No.	Cooling tower installations	Capacity of Cooling, m^3/hr	Service
1	Thermal Power Plant, Panki, Kanpur.	16,000	Power Plant
2	Durgapur Projects Ltd., Durgapur, W.B.	42,000 (3 units)	"
3	Talcher Thermal Scheme, Talcher, Orissa.	37,140	"
4	Ahmedabad Electricity Supply Co., Sabarmati.	26,450 (nat.draft, 15 units)	"
5	Andhra Pradesh Electricity Baard, Kathagudam.	21,400 (nat.draft, 3 units)	"
6	Renusagar Power Supply, Renusagar.	14,000	"
7	Hindustan Steel Ltd., Rourkela, Orissa.	50,000	Steel Plant
8	Durgapur Steel Plant, Durgapur, W.B.	19,550 (nat.draft)	"
9	Fertilizer Corpn.of India Ltd.,Gorakhpur.	21,250 (4 units)	Fertilizer
10	Neyvelli Lignate Corpn. Neyvelli, Madras.	80,000	Fertilizer and Power Plant

11	Gujarat State Fertilizer Co., Baroda.	13,000	Fertilizer
12	Indian Explosive Ltd., Panki, Kanpur.	13,800 (3 units)	"
13	Synthetics and Chemicals Ltd.,Bareille, U.P.	8,500	Synthetic rubber
14	National Organic Chemicals Corpn. Ltd., Bombay.	14,000	Petrochemicals
15	Indian Oil Corpn. Ltd., Kovali, Gujarat.	13,316	Oil-refiner
16	Worli Dairy Scheme, Bombay.	1,363	Dairy
17	Hindustan Photofilm Mfg. Co. Ltd., Ootacamund.	3,480	Photofilm

Many improvements have been made by the individual cooling tower companies on the basic theories used for analysis. Because the industry has a very competitive market, these refinements have been considered proprietory and closely guarded. Several papers have been written by the companies describing the theory but few publications are available concerning the practical applications.

The manufacturer is required to have a set of

guaranteed performance curves covering operating conditions such as water flow, cooling range, cold water temperature, and wet bulb temperature etc., in order to design a cooling tower. The cooling tower manufacturers in our country do not have guaranteed performance curves which are related to Indian conditions. These firms mainly depend upon their collaborators.

In the present work, known and proven theories are applied and computer programming has been developed for computing guaranteed performance curves for both counter flow and cross flow mechanical draft cooling towers for power plants, fertilizer and air conditioning plants for Indian conditions. Various major cities are considered in studying the performance curves of cooling towers with varying wet bulb temperatures since wet bulb condition of the entering air is one of the most important factors affecting the cooler performance.

The curves developed in the present work should be of great help to the cooling tower industry in the country and to the buyers in selecting and predicting tower performance at varying operating conditions.

CHAPTER II

FUNDAMENTALS OF COOLING TOWER ANALYSIS

The generally accepted concept of cooling tower performance was developed by Merkel [3] in 1925. His analysis and equations include the sensible and latent heat transfer into an overall heat and mass transfer process based on enthalpy difference as the basic driving force.

Consider a counter-flow tower of 1 m^2 ground area through which G kg/hr of air is flowing upward and L kg/hr of water is flowing downward. The counter-flow tower can be resolved into a one-dimensional problem [10] with the assumption that the flow pattern is vertical with the water falling downward through the tower and the air being forced upward.

Each water particle is surrounded by a film of saturated air at the bulk water temperature as shown in figure (2.1). The air is being heated and saturated as it passes through the tower. The heat is transferred from the interface to the main air mass by

(i) a transfer of sensible heat and

(ii) by the latent heat equivalent of the mass transfer resulting from the evaporation of a portion of the bulk water. The two processes are combined, into a single equation

$$L\ dT = K\ a\ dv\ (h''-h) = G\ dh \qquad (2.1)$$

where,

L = Water flow rate, kg/hr. m^2 of tower cross-section.

dT = Temperature differential, ^oC.

K = Overall mass transfer coefficient, kg/hr (m^2 of contact area) (kg water/kg dry air).

a = Interfacial contact surface, m^2/m^3 of tower volume.

dv = Differential volume, m^3.

h'' = Enthalpy of saturated air at water temperature, kcal/kg dry air.

h = Enthalpy of main air stream, kcal/kg dry air.

G = Air rate, kg/hr.m^2 of tower cross-section.

V = Tower volume, m^3/m^2 of tower cross section.

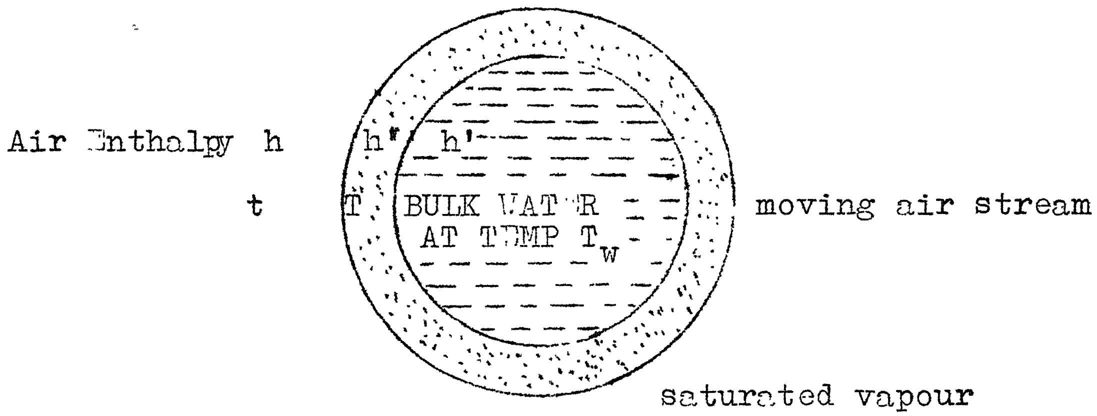


Fig. (2.1), Heat and Mass transfer between water, water vapour-film and air.

In equation (2.1), Merkel makes the following assumptions:

(a) The Lewis relationship $\frac{\alpha}{K.C_p} = 1.0$ (2.2)

where,

α = Coefficient of heat transfer by convection, kcal/hr.m^2 ^oC.

C_p = Specific heat of air, kcal/kg ^oC.

which holds true for the cooling tower (in the case of water evaporating into air, conductivity is approximately equal to diffusivity and hence this ratio is unity [11]).

(b) The air enthalpy can be expressed as

$$h = C_p\ T + h_{fg,s}W \tag{2.3}$$

where,

$h_{fg,s}$ = Latent heat of vaporisation for water, kcal/kg.

W = Humidity ratio of moist air, kg water vapour/kg dry air.

The equation (2.3) is an approximate one, which is derived by neglecting the superheat in the vapour and the heat of the liquid corresponding to the vapour content.

Integrating equation (2.1), we get

$$\int_{T_2}^{T_1} \frac{dT}{h''-h} = \frac{KaV}{L} \tag{2.5}$$

$$\text{or} \int_{h_1}^{h_2} \frac{dh}{h''-h} = \frac{KaV}{G} \tag{2.5'}$$

In arriving at equations (2.5) and (2.5)', the resistance to mass transfer from bulk water to interface, the temperature differential between the bulk water and interface and the effect of evaporation have been ignored. The left hand sides of the equations contain only the thermodynamic conditions for the cooling process. It is determined wholly by the initial and end conditions of the air flowing through the tower. The right hand sides of the equations are independent of the thermodynamic conditions in the tower and are determined by the characteristic of the tower design KaV and the water and air flow rates L and G.

Equations (2.5) and (2.5)' are the basic equations for calculating cooling tower performance.

The performance of a cooling tower quite patently depends upon a number of variables. The cooling tower industry today is called upon to design towers requiring large ranges and close approaches to the wet bulb.

In order to construct the performance curves it is of course, necessary to choose from among the many variables those which are to be used as co-ordinates, those to be used as parameters, and those to be held constant. This involves the following variables:

(i) approach to the wet bulb
(ii) range
(iii) L/G ratio
(iv) wet bulb temperature of the entering air.

The tower characteristic $\frac{KaV}{L}$ is the required factor and, therefore, it must be chosen as the ordinate. L/G is selected as abscissa, since it is one of the most important design characteristics.

In cooling tower practice, G is usually chosen from power requirement considerations. If L/G is known, then L can be obtained and therefore the necessary tower ground area for a given tower filling and a given tower capacity can be calculated.

Generally a cooling tower is selected initially by reference to sets of performance curves which each individual manufacturer has prepared for his own use, and which cover the types of tower and packings which are carried by that firm. Usually such diagrams consist of charts showing the approach to wet bulb temperature for different atmospheric conditions and varies as functions of the tower characteristic $\frac{KaV}{L}$ and $\frac{L}{G}$.

Thus for any wet bulb temperature, range and approach, and for any chosen $\frac{L}{G}$ ratio, these curves give

$$\int_{T_2}^{T_1} \frac{dT}{h''-h}$$

Since KaV is a function of L, as well as of G, for each tower design, curves of $\frac{KaV}{L}$ as functions of

$\frac{L}{G}$ should be constructed.

When analyzing the cooling tower performance, the application of the equations become extremely complex since, with the cooling tower, if one factor of the equation is changed, it starts a chain reaction, automatically changing other factors involved. Consequently, it becomes difficult to segregate and evaluate the effect of each individual factor.

The performance chart allows to select a cooling tower and predict its performance at any other conditions. It helps in evaluating the performance of an installed cooling tower. With one test taken at any operating conditions, it is possible to:

(a) determine if the tower is delivering its rated capacity,

(b) predict the tower's performance at any other operating conditions.

The performance of a cooling tower depends upon a number of variables. These are the cooling range, the approach to the wet bulb, the entering wet-bulb temperature, and the total circulating water flow rate. Performance is also affected by the tower and packing design and by the water-to-air ratio required to meet the specific design point.

<u>Packing</u>: The performance of a cooling tower is largely dependent upon the packing design. The function of

the fill is to increase [12] the contact between air and water by offering new exposed surfaces to the air as the water flows through the tower. Another function is to maintain proper distribution of both air and water.

The performance of a cooling tower is improved by increasing the amount of filling, height, area and/or air quantity. Increasing the tower height increases the length of time the air is in contact with the water, without seriously affecting the fan power required, but increases the pumping power. Increasing the tower area while maintaining the constant fan power increases the air quantity somewhat and increases the time that this air is in contact with the water because of lower velocity. The surface area of water in contact with the air is increased in both cases.

Considerable work has been reported in the literature about the use of various types of packings. Lichtenstein [13], London et al [14] and many other investigators have reported the performance characteristics of wood grids as cooling tower packings. Lowe and Christie [15] investigated the packing made with both flat and corrugated asbestos cement sheets. Narayankhedkar et al [16] investigated the packing consisted of alternately arranged flat and corrugated aluminium strips. This packing gave higher values of Ka under approximately same values of L and G compared to the packings used by other investigators [13, 15]. Bulanina et al [17] mainly concentrated on

fillings for the high capacity cooling towers rated at 65,000 - 100,000 m^3/hr. Aerodynamic and thermal conditions in cooling towers having a wetted area up to 12,000 m^2 were investigated. 3 types of fittings were considered. They were flat asbestos-cement panels, corrugated asbestos-cement panels, and wooden slots.

The most suitable arrangement of the packing can only be found by experiment. The object is to achieve a sufficiently large cooling surface whilst minimizing the resistance offered by the packing to the passage of air, a loss in the performance as a result of a slightly smaller cooling surface can be compensated by increasing the air flow and air velocity for the same height of stack. Increasing the air velocity through the tower decreases the time, the air is in contact with the water, but, since a greater quantity is passing through, the average differential between the water temperature and the wet bulb temperature of the air is increased, and this increases the rate of heat transfer. Increased air quantities are obtained only at the expense of increased fan power, which increases approximately as the cube of the fan speed [18].

When it is very difficult to determine accurately the free surface of the liquid, e.g., on breaking up the flow of circulating water into droplets, the volumetric of heat and mass transfer coefficients are used, i.e., coefficients that are based, not on the unit surface of the water, but on the unit active volume of the cooler [4].

L.D. Berman [4] found that the volumetric mass transfer coefficient Ka for a tower packing is approximately proportional to the mass velocity of air to the power 0.55 - 0.60 and to the superficial water flow rate to the power 0.3 - 0.4. These experiments are also confirmed by Lichtenstein.

For a counterflow tower with a packing of rectangular slats Lichtenstein [13] obtained an empirical relation:

$$Ka = A\, G^{m}\, L^{n} = B\, G^{m}\, l^{n} \tag{2.7}$$

The constants in this equation are:

A = 635, B = 1050

m = 0.53 and n = 0.39.

The size and arrangement of the slat packing are given in figure (2.2).

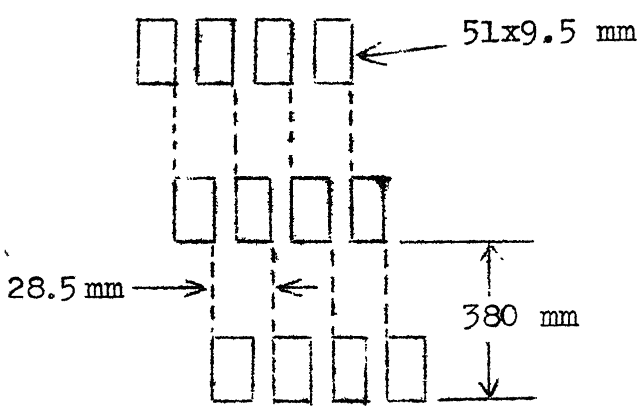


Fig. (2.2), Size and arrangement of rectangular slats by Lichtenstein.

The volumetric mass transfer coefficient for the above tower packing as a function of water flow rate is given in Table (2.1).

Table (2.1), volumetric mass transfer coefficient for the packing in figure (2.2) as a function of water flow rate.

Water flow rate $L (kg/hr.m^2)$	Ka for $G=3000 (kg/hr.m^2)$	Ka for $G=6000 (kg/hr.m^2)$	Ka for $G=8000 (kg/hr.m^2)$
4,000	1,050	1,450	1,750
6,000	1,200	1,700	2,050
8,000	1,400	1,950	2,300
10,000	1,575	2,150	2,530
12,000	1,700	2,300	2,700

Water Load: For any industrial plant, the method of heat dissipation dictates the amount of water flow through the intake. The amount of water passing through the cooling tower, L, is determined by the amount of heat to be transferred and the allowable increase in the cooling water temperature:

$$L = \frac{Q}{C_w \cdot \Delta T} \qquad (2.6)$$

where,

Q = Heat removed per unit time, kcal/hr.

T = Allowable temperature increase of cooling water, ^{o}C.

C_W = Heat capacity of cooling water, kcal/kg.^{o}C.

For a cooling tower, if the mass air rate remains constant, an increase in water rate results in a higher wet bulb temperature of the outgoing air. The temperature of the air leaving the tower can approach, but not exceed, the temperature of the water entering the tower.

Investigators have placed an upper and lower limit on water loading, some even stating that a loading between 5.0 and 12.0 $m^3/hr.m^2$ gives optimum [19] performance. The reason is that a light loading results in poor water distribution and unequal wetting of the filling while a high loading causes what is referred to as flooding.

Capacity of an existing tower cannot be increased by pouring more water over it. Additional water still contacts the same amount of air, with the net effect of raising the cold water temperature. It is not advisable to reduce water flow when less output is needed. Output should be decreased by shutting down cells, reducing fan speed, or turning off fans.

Evaporative water losses may vary widely, depending upon the type of cooling facilities and the existing ambient conditions.

The general water balance assessment of a cooling system is expressed as :

$$M = E + D + B \tag{2.8}$$

where,

M = Make up water flow rate,

E = evaporation rate,

D = drift and windage loss rate, and

B = blowdown flow rate.

Evaporation loss averages approximately 1 per cent of the circulating water for every $5^{\circ}C$ cooling range [20]. The drift loss on a modern mechanical draft tower is considered to be something less than 0.1 per cent [21] of the circulating water flow rate.

<u>Cooling Range, Approach and WBT</u>: These affect the tower performance in terms of the tower size and cost. The broader the range, the more expensive the tower is. Longer air-water contact, which brings about greater temperature drops, depends on a larger fill area and fan. Thus a broad range will also result in an increased cost of operation.

The smaller the approach, the higher the cost of the tower. Trying to lower water temperature by 1 or 2 extra degrees requires dramatic increases in tower size and operating costs.

The design wet bulb temperature should be selected on economical conditions [22] and will not necessarily be the highest WBT registered in the area.

As the periods when the actual wet bulb temp-

erature exceeds the design WBT are for short duration during summer mid-day hours, it is more economical to operate at slightly higher water temperature from the tower during these hours than to install a tower that would be oversize for all but these short periods.

It is characteristic for natural draft towers to have relatively better output at lower wet bulb and for mechanical draft towers to have better performance at higher wet bulbs.

Methods To Evaluate Tower Performance:

COUNTER FLOW TOWER PERFORMANCE: The method of correlation developed by Lichtenstein [13] can be used for designing, evaluating and predicting counterflow tower performance. Experience and research conducted by the cooling tower industry and other organisations have shown that this method approximates the performance of counterflow cooling towers. The method has gained wide acceptance by the industry.

A graphic representation of $\int_{T_2}^{T_1} \frac{dT}{h''-h}$ is shown in figure (2.3).

The water temperature T is selected as abscissa and the enthalpy as the ordinate. Saturation line h'' gives the enthalpies of saturated air at water temperatures. In figure (2.3) area represented by ABCD determines the tower characteristic necessary to cool water from T_1 to T_2 with inlet air enthalpy of h_1 and a given water to air flow rate ratio L/G. The air

enthalpy varies as a straight line with the water temperature since L dT = G dh.

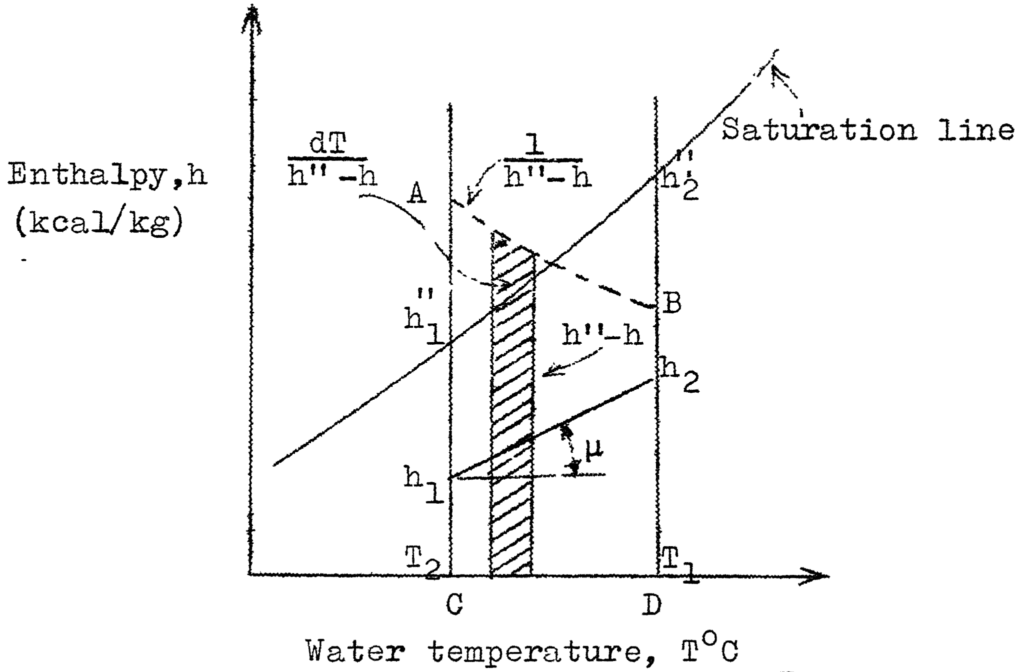


Fig. (2.3) Graphic representation of $\int_{T_2}^{T_1} \frac{dT}{h''-h}$

It is possible to introduce a ''log-mean potential'' as an approximate method of solution in accordance with the analogy of the heat exchanger. The log-mean enthalpy potential would be mathematically correct if the enthalpy potential h''-h were a straight function of T. This is true only if the saturation line is straight, and so it follows that the log-mean potential can give good results only for very small ranges over which the saturation line could be considered as

approximately straight. Test data taken under one set of conditions cannot be used accurately to predict performance under other conditions if this method is employed, since the error changes with the curvature of the saturation line. For practical cooling tower work, the use of a log-mean potential is, therefore, not adequate.

London et al [14] expressed the tower performance in terms of its effectiveness as an energy exchanger.

$$\varepsilon_h = \frac{L\ C_w(T_1-T_2)}{G\ (h_{a1}-h_2)} \tag{2.9}$$

where,

C_w = Unit heat capacity of liquid water, kcal/kg°C.

h_a = Enthalpy of the air-water vapour mixture at the equilibrium WBT, kcal/kg.

h = Enthalpy of air stream, kcal/kg.

Suffix, 1 = Water entrance, air exit.

Suffix, 2 = Water exit, air entrance.

If, as normally is the case, the cooling range is such that

$$T_2 \gg t_{wb2},\ \text{then}\ L\ C_w(T_1-T_2) \simeq G\ (h_1-h_2)$$

$$\therefore\ \varepsilon_h = \frac{G(h_1-h_2)}{G(h_{a1}-h_2)}$$

$$= \frac{h_1-h_2}{h_{a1}-h_2} \tag{2.10}$$

i.e., ε_h represents the ratio of actual tower energy exchange between phases to the energy exchange which

would result provided the discharged air was saturated at the temperature of the entering water.

ε_h may be represented empirically by an exponential equation:

$$\varepsilon_h = 1 - C\, e^{\frac{-\eta K a V}{G}} \qquad (2.11)$$

where,

C = Dimensionless Coefficient.

η = Fraction of the packing area aV covered by the flowing water.

In this method, C and η in equation (2.11) had to be found experimentally. The effectiveness was found to vary from 0.3 at low water rates and high air rates, to 0.8 at high water rates and low air rates.

Baker et al [19] presented the Unit-Volume Coefficient as a method of cooling tower performance. In this method the integration is accomplished by increments of equal volumes. The Unit-Volume Coefficient can be defined as kcal transferred per m^3 of tower per m^2 of plan area per kcal difference in enthalpy potential. In this method, it is not possible to start with the coefficient and solve for the predicted performance conditions except by trial and error.

Hallett [23] with the help of CTI (Cooling Tower Institute) bulletins ATP-107R and ATP-127, presents methods which are used for calculating performance of a mechanical draft cooling tower. In this method, the

curves may be calculated using analytical methods similar to those used by the Cooling Tower Institute to determine performance levels from field test data. A format and calculation procedure for computing $\frac{K\,a\,V}{L}$ based on Tchebyshev's method for numerically evaluating the integral (eqn.2.5) is shown in detail in Chapter III. This method is employed in the present work in evaluating the performance curves since this method gives consistent results over a wide variety of cooling ranges and wet bulb temperatures. The method also imparts itself to programming on a digital computer.

<u>CROSS FLOW TOWER PERFORMANCE</u>: References on the <u>cross flow principle</u> of water cooling originated in the early 1920's. Some of the first cooling devices such as spray ponds and towers incorporated the cross flow principle in part, but the process was not analysed separately.

The greatest advantages of the industrial cross flow tower are its design capability of water loadings to 50 $m^3/hr.m^2$ of packing area (20 gpm/ft^2 of packing area) and its air velocities to 3.0 m/sec (600 fpm).

The methods used for estimating performance of counter flow towers cannot be applied to cross flow towers with the same degree of accuracy, although there are certain similarities in the methods. The heat and mass transfer processes in both types of towers are based on the same potential for cooling.

The method of analysis and the process of integration are dependent upon the relative flow pattern of water and air in each type of tower.

The cross flow tower involves a two-dimensional flow pattern in which water falls downward through the tower and the air is drawn horizontally through the packing. The enthalpy of the air changes not only in the vertical direction but also changes in the horizontal direction.

In 1956, Zivi and Brand [24] presented a method for cross flow tower analysis based on the principle of enthalpy differential as the potential. They considered a vertical section through a cross flow cooling tower as shown in fig. (2.4).

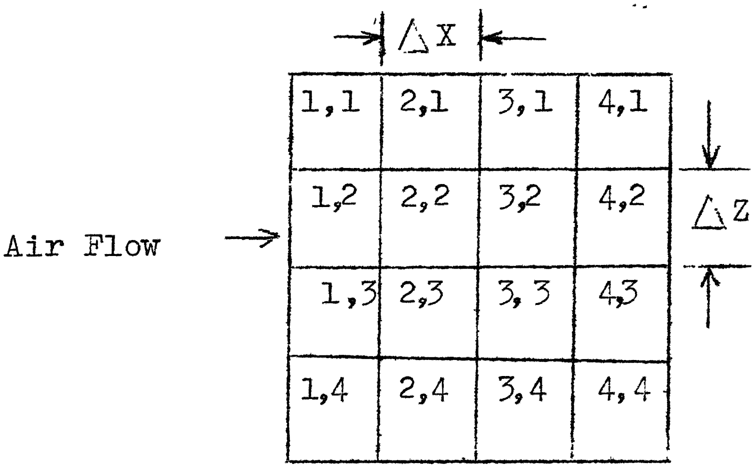


Fig. (2.4) Vertical section through a cross flow cooling tower.

The width of the tower is taken as unity. Hori-

zontal air flow is induced by the fan. The water enters at the top and flows uniformly downward through the fill. The positive X-direction is defined as the direction of the air flow, and the positive Z-direction as the direction of the water flow. The water temperature entering the top of the fill is at the same temperature along the entire length. As the water flows downward, it is cooled. The water in the left hand portion of the fill is exposed to cooler and drier air than is the water in the right hand portion. Therefore, the rate of cooling of the water on the left is greater than that on the right.

Thus, their method consists of a point-by-point determination of the water temperature and air enthalpy distribution by considering a small element of volume of the fill, and writing the equations of energy conservation and heat transfer. The contours of constant average water temperatures are plotted on coordinates of non-dimensional tower height and depth. The dimensionless curves can be used to determine the characteristic required for a cross flow tower to meet a given cooling specification.

Vouyoucalos [25] analysed the cross flow cooling tower by considering an element of volume with finite dimensions but sufficiently small so that the relation $h'' = f(T)$ could be approximated by a straight line.

$$h'' = mT + p \tag{2.12}$$

where,

m = Constant, kcal/kg$^{\circ}$C.

p = Constant, kcal/kg.

Considering enthalpy transfer equation:

$$L\ C_w dT = K\ a\ (h''-h)\ dZ \qquad (2.13)$$

$$\therefore\ Z = \frac{L}{K\ a}\int_1^2 \frac{dT}{h''-h} \qquad (2.14)$$

where,

Z = Side of the volume element, in meters.

1,2 = Bottom and top of the tower.

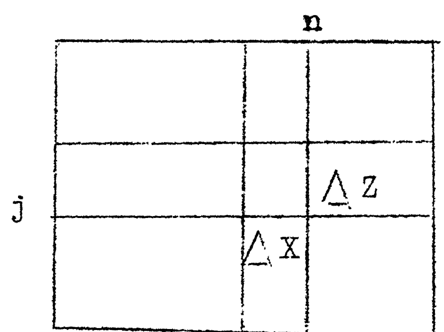


Fig. (2.5) Tower co-ordinates ($\triangle X = \triangle Z$)

For this volume figure (2.5), equation (2.14) can be applied. By combining equation (2.14) with equation (2.12) and integrating

$$\frac{\triangle Z\ K\ a}{L} = \frac{(T_{j-1,n} - T_{j,n})\ \ln \dfrac{(h''-h)_{j,n-1}}{(h''-h)_{j,n}}}{(h''-h)_{j,n-1} - (h''-h)_{j,n}} \qquad (2.15)$$

Since the variation of the group $(h''-h)$ is not very great within a volume, the log-mean can be approximated to the arithmetic mean

$$\therefore \frac{\Delta Z\ K\ a}{L} = \frac{2(T_{j-1,n} - T_{j,n})}{(h''-h)_{j,n-1} + (h''-h)_{j,n}} \qquad (2.16)$$

Considering the variation of h'' within a volume element is not great

$$(h'')_{j,n-1} \simeq (h'')_{j,n}$$

$$\therefore \frac{\Delta Z\ K\ a}{L} = \frac{2(T_{j-1,n} - T_{j,n})}{2h''_{j,n-1} - h_{j,n-1} - h_{j,n}} \qquad (2.17)$$

By (2.12),

$$h''_{j,n-1} = m\ T_{j-1,n} + p \qquad (2.18)$$

By basic equation $L\ dT = G\ dh$

$$h_{j,n} = (T_{j-1,n} - T_{j,n})\ \frac{L}{G} + h_{j,n-1} \qquad (2.19)$$

By adding (2.17), (2.18) and (2.19)

$$T_{j,n} = B\ T_{j-1,n} + C\ h_{j,n-1} + D \qquad (2.20)$$

where,

j = rows

n = columns

$$B = \frac{\left[\dfrac{K\ a\ \Delta Z}{2G} - \dfrac{K\ a\ \Delta Z\ m}{L} + 1\right]}{\left[\dfrac{K\ a\ \Delta Z}{2G} + 1\right]} \text{ used in (2.20)}$$

$$C = \frac{\left[\frac{K\ a\ \Delta Z}{L}\right]}{\left[\frac{K\ a\ \Delta Z}{2G} + 1\right]} \quad \text{used in (2.20)}$$

$$D = \frac{\left[\frac{p\ K\ a\ \Delta Z}{L}\right]}{\left[\frac{K\ a\ \Delta Z}{2G} + 1\right]} \quad \text{used in (2.20)}$$

The enthalpy of the incoming air was $h_{j,o}$ and the temperature of the incoming water was $T_{o,n}$, both known. Equations (2.19) and (2.20) can be used to find the enthalpy of the air and temperature of the water in any part of the tower, starting from the volume element (0,0).

This method could not be adopted in drawing the cross flow tower performance curves in the present work since, many variables were unknown and had to be obtained experimentally.

Baker and Mart [19] applied the same analysis of Unit-Volume Coefficient to cross flow tower although the calculations are more tedious. The mechanical integration was accomplished by dividing the cross section in a number of columns, each of which is subdivided in a series of incremental volumes. The mechanical integration of the coefficient for set of performance conditions is not too time consuming, but it is not possible to start with the coefficient and

solve for the predicted performance conditions except by trial and error.

Hallett [23] modified the Zivi and Brand method and showed the effect of variables which had been discussed but not completely defined by previous authors. The equations have been given in a general form. The performance curves are dimensionless and are not related to a particular tower design. Hence, this method has been employed in the present work in drawing the performance curves of cross flow cooling tower. The format and calculation procedure to evaluate the enthalpy of air and temperature of water with the equations is shown in detail in chapter III. This method also contributes itself to programming on a digital computer.

CHAPTER III

PERFORMANCE CURVES EVALUATION

Various methods have been described in the previous chapter to compute the performance curves for the mechanical draft cooling towers. In the present chapter, the methods proposed by Zivi and Brand [^4] and Hallett [23] have been adopted for computing performance curves for the counter flow and cross flow towers, respectively. Computer programs have been developed using IBM 7044/1401 computers. The following industries have been taken into account as representative users of cooling towers:

(1) Thermal power plants,
(2) fertilizer plants and
(3) air conditioning plants.

Tower performance has been studied for various design wet bulb and dry bulb temperatures for various major cities of India. The design values of wet bulb temperature and dry bulb temperature have been supplied by Paharpur Cooling Towers Pvt. Ltd., Calcutta as listed in table (3.1).

Table (3.1), Design wet bulb and dry bulb temperatures for various locations in India.

Sl. No.	City	Lat. °N	Deviation of I.S.T. from Solar noon, mins.	Design WBT, °C	Design DBT, °C
1	Ahmedabad	23	+41	27.8	41.1
2	Allahabad	25	+23	28.3	41.5
3	Amritsar	32	+32	28.3	40.0
4	Asansol	24	-10	28.3	40.0
5	Bombay	19	+41	27.8	33.9
6	Calcutta	23	-2[illegible]	28.3	36.7
7	Delhi	29	+23	28.3	40.5
8	Gauhati	26	-35	28.3	30.5
9	Gaya	25	- 8	28.3	41.7
10	Gwalior	26	+19	27.8	41.6
11	Jamshedpur	23	-13	28.3	40.5
12	Jaipur	27	+29	26.6	40.0
13	Jodhpur	26	+39	27.8	41.5
14	Kanpur	27	+10	28.3	41.1
15	Lucknow	27	+ 8	28.3	40.0
16	Madras	13	+11	28.3	37.2
17	Nagpur	21	+15	26.1	42.6
18	Patna	26	- 9	28.3	39.4
19	Poona	18	+36	24.5	38.3
20	Trivendrum	8	+24	27.2	32.7
21	Vishakapatnam	18	- 2	28.9	36.6

<u>COUNTER FLOW PERFORMANCE CURVES</u>: The curves for counter flow cooling tower are computed by using the Tchebyshev's method proposed by the Cooling Tower Institute [26] and

by Hallett [23].

The calculation procedure is explained as following:

In order to determine the cooling tower characteristic $\frac{K\,a\,V}{L}$, it is necessary to integrate the definite integral (eqn. 2.5).

$$\int_{T_2}^{T_1} \frac{dT}{h''-h}$$

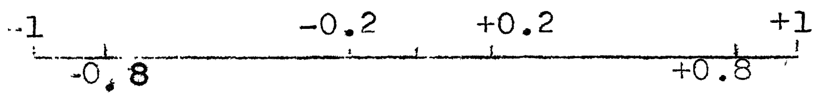
or

$$\int_{T_2}^{T_1} \frac{dT}{\Delta h} \tag{3.1}$$

between the known limits of water temperatures T_1 and T_2. The Tchebyshev's method of integration is used according to which:

$$\int_{-1}^{1} f(x)\,dx \approx \frac{2}{n} \sum_{i=1}^{n} f(x_i) \tag{3.2}$$

where x_i, indicate the real roots of the Tchebyshev quadrature polynomial for different values of n. For n = 4, the roots x_i are given as $\pm$ 0.187592 $\approx \pm$ 0.2 and $\pm$ 0.794654 $\approx \pm$ 0.8.



Now, in the integral (3.1), we can express

$$T = \frac{T_1 + T_2}{2} + \frac{T_1 - T_2}{2}\,\delta$$

when $\delta = 1$, $T = T_1$

when $\delta = -1$, $T = T_2$

Also $dT = \frac{T_1 - T_2}{2}\, d\delta$

Substituting T in terms of δ in the integral (3.1) and inserting the corresponding limits for δ, we get:

$$I = \frac{K\,a\,V}{L} = \int_{-1}^{1} \frac{\frac{T_1 - T_2}{2}\, d\delta}{\Delta h}$$

$$= \frac{T_1 - T_2}{2} \int_{-1}^{1} \frac{d\delta}{\Delta h}$$

$$= \left(\frac{T_1 - T_2}{2}\right) \frac{2}{n} \sum_{i=1}^{n} f(x_i), \text{using } (3.?)$$

for $n = 4$,

$$I = \frac{K\,a\,V}{L} = \left(\frac{T_1 - T_2}{2}\right)\frac{1}{2} \sum_{i=1}^{4} f(x_i)$$

$$= \left(\frac{T_1 - T_2}{4}\right) \sum_{i=1}^{4} \frac{1}{\Delta h_i} \qquad (3.3)$$

Consider the height of a counter flow tower such that T_1 and T_2 represent the temperatures of the inlet and the outlet water, as shown in the figure (3.1). These temperatures correspond to the points + 1 and − 1,

respectively of the interval (-1, + 1) considered in the integral (3.2).

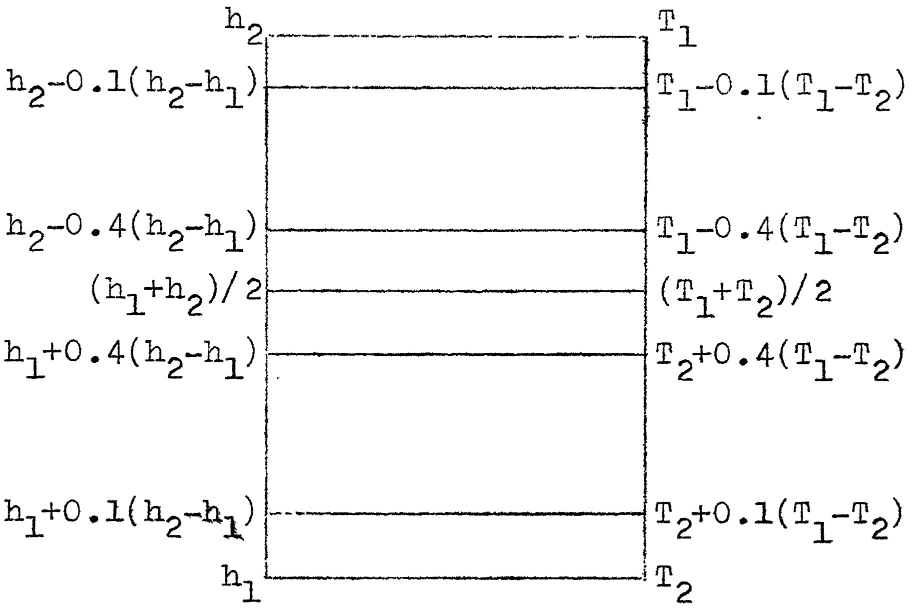


Fig. (3.1), nthalpy and temperature distribution in a counter flow tower.

In order, now, to determine the temperatures at the four specified points in the interval (-1,+1), we use the relation

$$T = \frac{T_1 + T_2}{2} + \left(\frac{T_1 - T_2}{2}\right) \delta$$

when $\delta = -0.8$,

$$T = 0.5(T_1+T_2)-0.4(T_1-T_2)$$

$$= T_2+0.1(T_1-T_2) \qquad (3.4)$$

when $\delta = -0.2$,

$$T = 0.5\ (T_1+T_2) - 0.1\ (T_1-T_2)$$

$$= T_2 + 0.4\ (T_1-T_2) \tag{3.5}$$

when $\delta = +0.2$,

$$T = T_1 - 0.4\ (T_1-T_2) \tag{3.6}$$

when $\delta = +0.8$,

$$T = T_1-0.1\ (T_1-T_2) \tag{3.7}$$

The relations (3.4 - 3.7) represent the temperatures at four different sections of a counter flow cooling tower.

Similarly, we can express the enthalpy as:

$$h = \frac{h_1 + h_2}{2} + \frac{(h_2 - h_1)}{2}\delta \tag{3.8}$$

Following the same procedure, the enthalpy at the same corresponding sections of the tower can be obtained as:

$$h = h_1 + 0.1\ (h_2-h_1) \tag{3.9}$$

$$h = h_1 + 0.4\ (h_2-h_1) \tag{3.10}$$

$$h = h_2 - 0.4\ (h_2-h_1) \tag{3.11}$$

$$h = h_2 - 0.1\ (h_2-h_1) \tag{3.12}$$

These results are arranged in table (3.2) in a format used for our calculation procedure.

Table (3.2), Calculation procedure for temperature and enthalpy in a counter flow tower.

Water Temp., ^{o}C	Enthalpy of sat. air at water temp., h''(kcal/kg dry air)	Enthalpy of air at air temp., h(kcal/kg dry air)	$h''-h=\Delta h$	$\frac{1}{\Delta h}$	$\sum\frac{1}{\Delta h}$
$T_2=...$		$h_1=....$			
$T_2+0.1(T_1-T_2)$ =.....	....	$h_1+0.1(h_2-h_1)=....$	....	...	...
$T_2+0.4(T_1-T_2)$ =.....	....	$h_1+0.4(h_2-h_1)=....$	....	...	...
$T_1-0.4(T_1-T_2)$ =.....	....	$h_2-0.4(h_2-h_1)=....$	....	...	...
$T_1-0.1(T_1-T_2)$ =.....	....	$h_2-0.1(h_2-h_1)=....$	....	...	...
$T_1=...$		$h_2=....$			

$$\therefore \int_{T_2}^{T_1} \frac{dT}{h''-h} = \frac{T_1-T_2}{4} \sum_{i=1}^{4} \frac{1}{\Delta h_i}$$

The design data on cooling towers for different services have been collected from various cooling tower installations in Kanpur city. They are shown in table (3.3), which consists of design values of water circulation rate (L), air flow rate (G), inlet water temperature (T_1), outlet water temperature (T_2), inlet wet bulb temperature (t_{wb1}) etc.

Using the known data from table (3.3), and adopting the procedure indicated in table (3.2), the value of the tower characteristic $\frac{K\,a\,V}{L}$ is obtained.

<u>Example</u>: Consider a counter flow cooling tower installation at Kanpur city for a thermal power plant of capacity 64 MW. The total heat to be dissipated is 128×10^6 kcal/hr. The inlet hot water temperature and outlet cold water temperatures are 43°C and 35°C, respectively (table 3.3). Hence, the cooling range is 8°C. The air flow rate is given as 8.66×10^6 kg/hr. The ambient air design wet bulb temperature for Kanpur is 28.3°C (table 3.1).

The following procedure is adopted to compute the design value of $\frac{K\,a\,V}{L}$.

$$Q = L \times A \times \Delta T \times C_w$$

where Q = Quantity of heat to be dissipated, kcal/hr.

ΔT = Cooling range, °C.

A = Area of cross-section of the tower = 1100 m^2.

C_w = Specific heat of water = 1 kcal/kg°C.

$$\therefore \quad L = \frac{Q}{\Delta T x A x C_w}$$

Table (3.3), Design Data

Sl. No.	Particulars	Power Plants		Fertilizer Plants			Air-conditioning Plant
		64MW	220MW	1	2	3	
1	No. of cells	6	6	3	3	1	1
2	Water circulation rate, L kg/hr	16×10^6	16.8×10^6	5.1×10^6	6.9×10^6	1.8×10^6	$.45 \times 10^6$
3	Inlet water temp., ^oC	43.0	43.0	45.2	42.0	42.5	35.55
4	Outlet water temp., ^oC	35.0	33.0	33.2	33.2	33.2	31.11
5	Cooling range, ^oC	8.0	10.0	12.0	8.8	9.3	4.44
6	Inlet wet bulb temp., ^oC	28.3	28.3	28.3	28.3	28.3	28.3
7	Approach, ^oC	6.7	4.7	4.9	4.9	4.9	2.8
8	Total heat to be dissipated, kcal/hr	128×10^6	168×10^6	61.2×10^6	60.72×10^6	16.74×10^6	2.0×10^6
9	Air flow rate, kg/hr	8.66×10^6	10.13×10^6	3.99×10^6	4.50×10^6	1.28×10^6	0.25×10^6

$\therefore$ water circulation rate $L = \dfrac{128 \times 10^6}{(43-35) \times 1100 \times 1}$

$= 14{,}545$ kg/hr.m^2

Air flow rate $G = \dfrac{8.66 \times 10^6}{1100}$

$= 7872$ kg/hr.m^2

$\therefore \quad \dfrac{L}{G} = 1.8467$

Inlet water temperature $T_1 = 43^\circ C$

Outlet water temperature $T_2 = 35^\circ C$

$\therefore$ Cooling range $(T_1-T_2) = 8^\circ C$

$\therefore \quad (h_2-h_1) = (T_1-T_2)\dfrac{L}{G}$

$= 1.8467 \times 8$

$= 14.7735$ kcal/kg dry air.

The enthalpy of air at the entering wet bulb temperature $28.3^\circ C$ = 21.801 kcal/kg dry air (from standard tables).

$h_1 = 21.801$ kcal/kg dry air

$h_2 = h_1 + (T_1-T_2)\dfrac{L}{G}$

$= 21.801 + 14.7735$

$= 36.5745$ kcal/kg dry air.

Using the table (3.2), the design value of the tower characteristic, $\dfrac{K\,a\,V}{L} = 0.9052$.

Corresponding to the design value of $\dfrac{K\,a\,V}{L}$ for a specified data, there is a design point P_D for the parameters specified in table (3.3). This is shown in figure (3.2). For the same value of the $\frac{L}{G}$ ratio

Water Temp., °C	Enthalpy of sat. air at water temp., h"(kcal/kg dry air)	Enthalpy of air at air temp., h(kcal/kg dry air)	$h'-h=\Delta h$	$\frac{1}{\Delta h}$	$\frac{1}{\Sigma h}$
$T_2=35.0$		$h_1=21.801$			
$T_2+0.1(T_1-T_2)$ $=35.8$	32.13	$h_1+0.1(h_2-h_1)$ $=23.278$	8.851	0.1129	
$T_2+0.4(T_1-T_2)$ $=38.2$	36.29	$h_1+0.4(h_2-h_1)$ $=27.710$	8.579	0.1165	
$T_1-0.4(T_1-T_2)$ $=39.8$	39.24	$h_2-0.4(h_2-h_1)$ $=30.665$	8.674	0.1152	
$T_1-0.1(T_1-T_2)$ $=42.2$	44.37	$h_2-0.1(h_2-h_1)$ $=35.097$	9.272	0.1078	
		$h_2=36.5745$			.4526

∴ The design value of the tower characteristic,

$$\frac{K\,a\,V}{L} = \frac{(43-35)}{4}\,(0.4526) = 0.9052$$

and the range, there may be many combinations of the wet bulb and the cold water temperatures to give the same design value of the tower characteristic $\frac{K\,a\,V}{L}$. For each such combination, there should be a corresponding point in the chart. By joining all these points including the "design point" we get a curve which is termed as the <u>Performance Curve</u>. Similar performance curves are obtained for many other cooling ranges. The following procedure is adopted: Estimate a cold water temperature or approach for the wet bulb temperature being considered. Using table (3.2), the value of $\frac{K\,a\,V}{L}$ is obtained for the estimated cold water temperature. The calculated value of $\frac{K\,a\,V}{L}$ is compared with the design value of $\frac{K\,a\,V}{L}$. If it does not agree within a reasonable tolerance, use a trial and error solution and a refined estimate for the cold water temperature until the tolerance of $\frac{K\,a\,V}{L}$ is met. The process is similarly repeated for different cold water temperatures.

To avoid the tedious calculations by hand, a computer program has been developed (Appendix-A) for following computation ranges covering all possible cooling tower installations in India.

(i) Wet bulb temperature : $15^{\circ}C - 30^{\circ}C$ by an increment of $1^{\circ}C$.

(ii) Cold water temperature : $15^{\circ}C - 40^{\circ}C$ by an increment of $1^{\circ}C$.

(iii) Cooling range : $1^{\circ}C - 20^{\circ}C$ by an increment of $1^{\circ}C$.

The program reads the above input data together

with the constant L/G ratio and the design value of $\frac{K\,a\,V}{L}$. A tolerance is given to get the accuracy of computation. The computation time required for calculating $\frac{K\,a\,V}{L}$ for the above ranges was 3 - 4 minutes on an IBM 7044/1401 computer. The subroutine ALPHA calculates the value of h'' for a particular value of T, to be used in table (3.2).

In plotting the curves, the wet bulb temperature is taken as abscissa and the cold water temperature as the ordinate. Cooling range was kept constant in drawing each curve. Figures (3.2) - (3.9) show sets of performance curves for various cooling ranges and L/G ratios for different applications.

Situation may, however, occur when it is necessary to determine the tower characteristic for various L/G ratios and approach values for a particular design wet bulb temperature and a fixed cooling range. The procedure adopted to draw the performance curves under these conditions is described below:

In this case, the cooling tower characteristic $\frac{K\,a\,V}{L}$ is computed for a series of L/G ratios and cold water temperatures. In drawing a chart, the wet bulb temperature and the cooling range are kept constant. $\frac{K\,a\,V}{L}$ is obtained by using the table (3.2). The following ranges were considered in order that the performance curves are applicable to the particular industrial locations of the country.

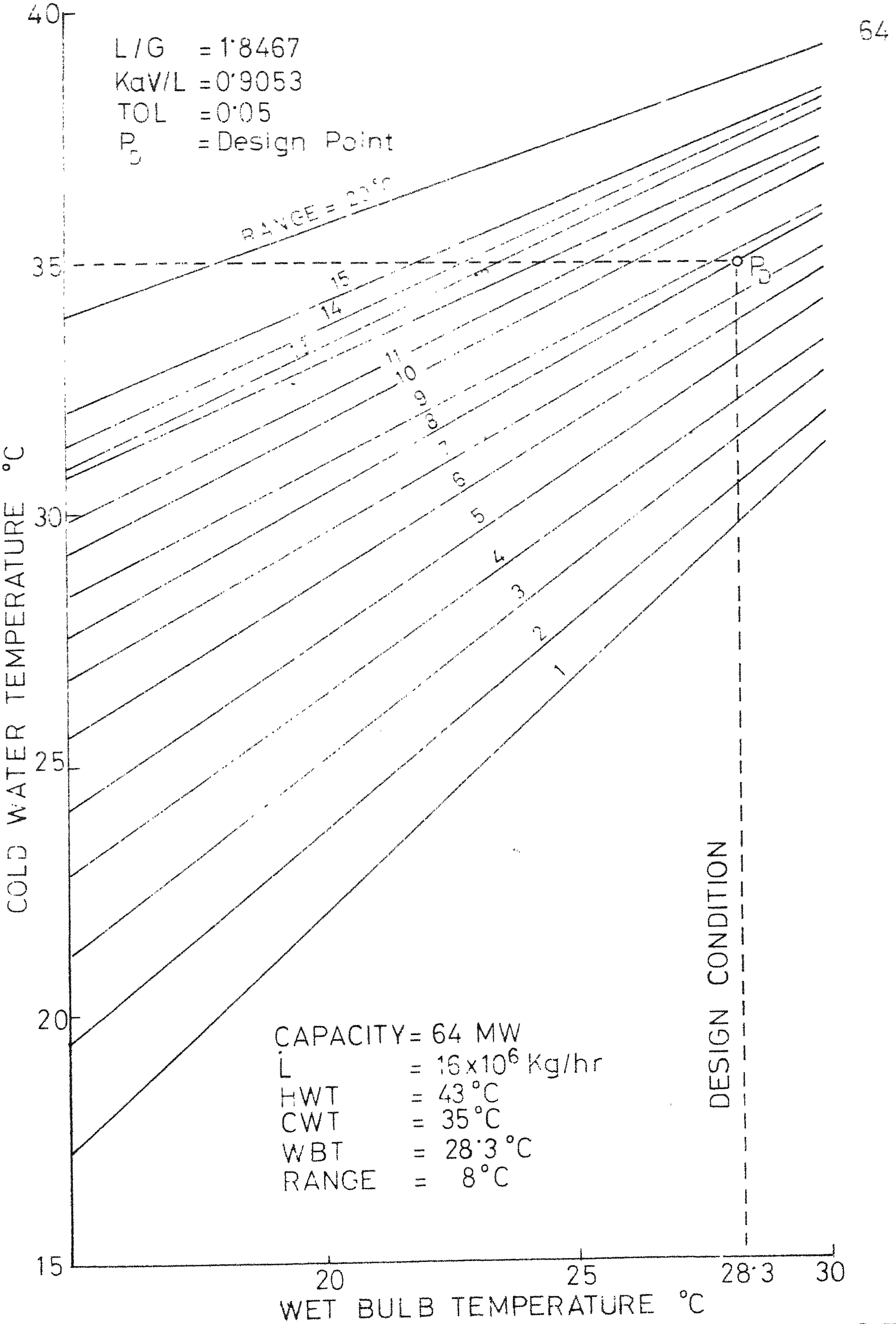


FIG. 3·2 COUNTER FLOW TOWER PERFORMANCE CURVES (THERMAL POWER PLANT).

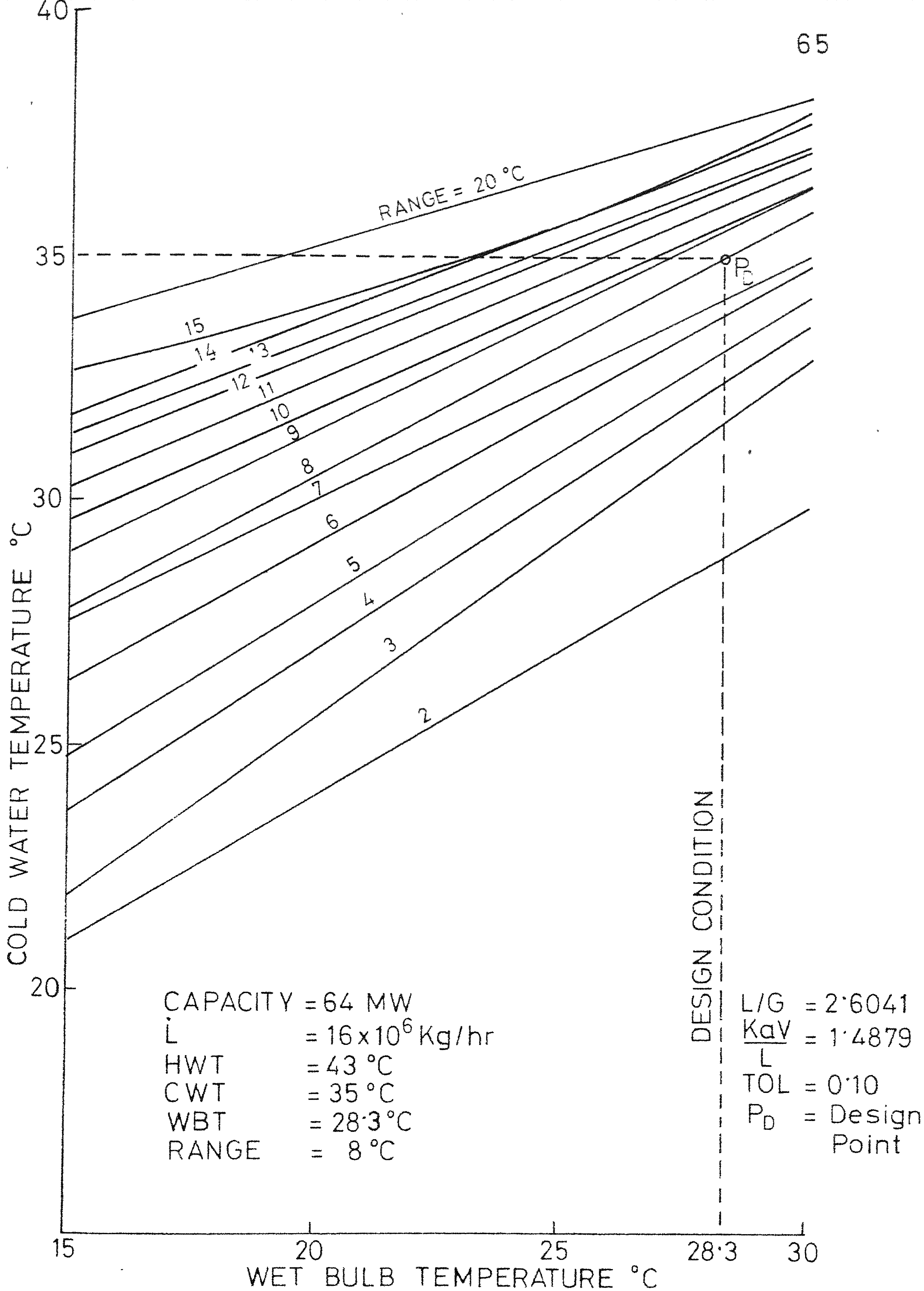


FIG. 3·3 COUNTER FLOW TOWER PERFORMANCE CURVES (THEMAL POWER PLANT).

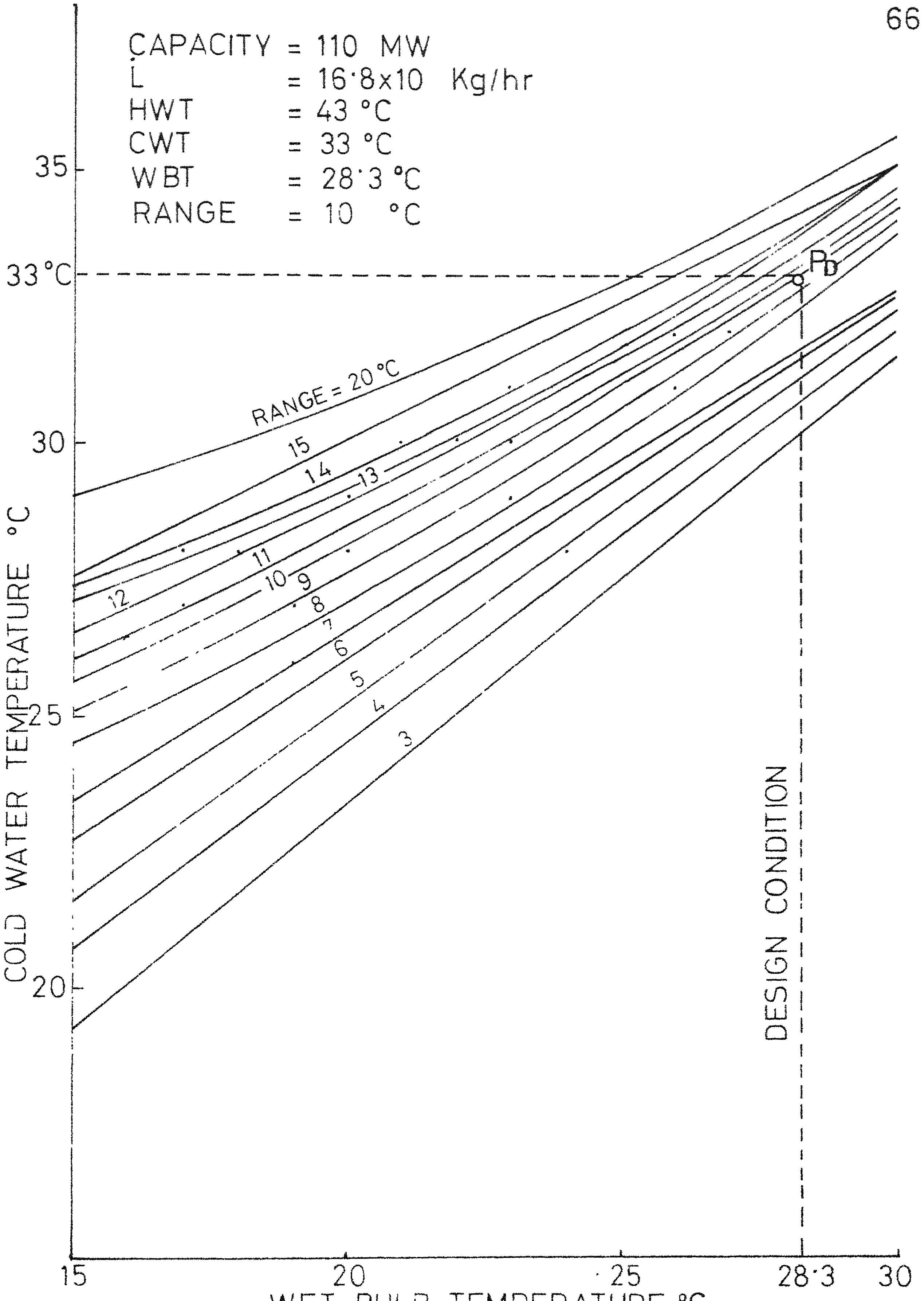


FIG. 3·4 COUNTER FLOW TOWER PERFORMANCE CURVES.
(THERMAL POWER PLANT)

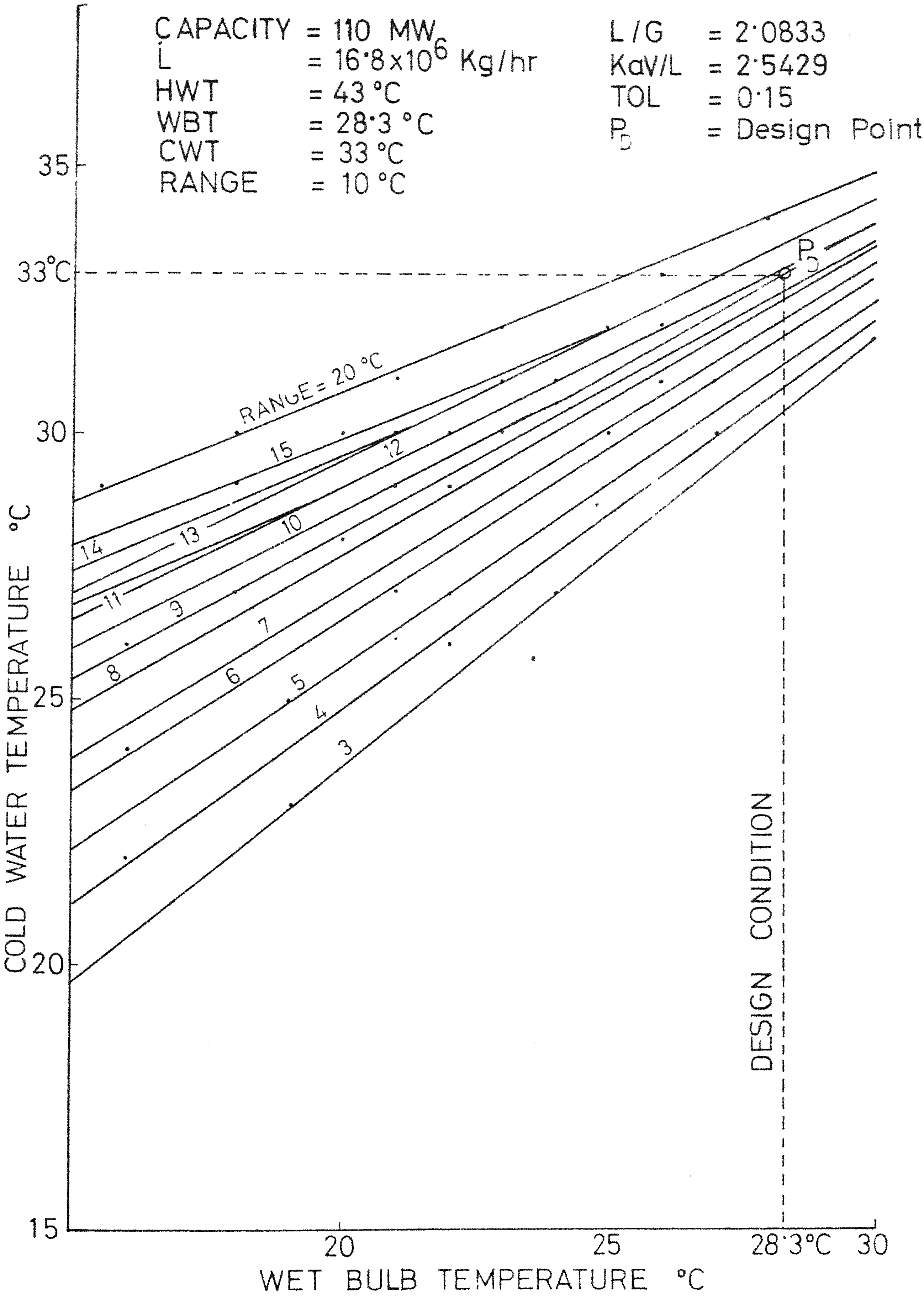


FIG. 3·5 COUNTER FLOW TOWER PERFORMANCE CURVES.

(THERMAL POWER PLANT)

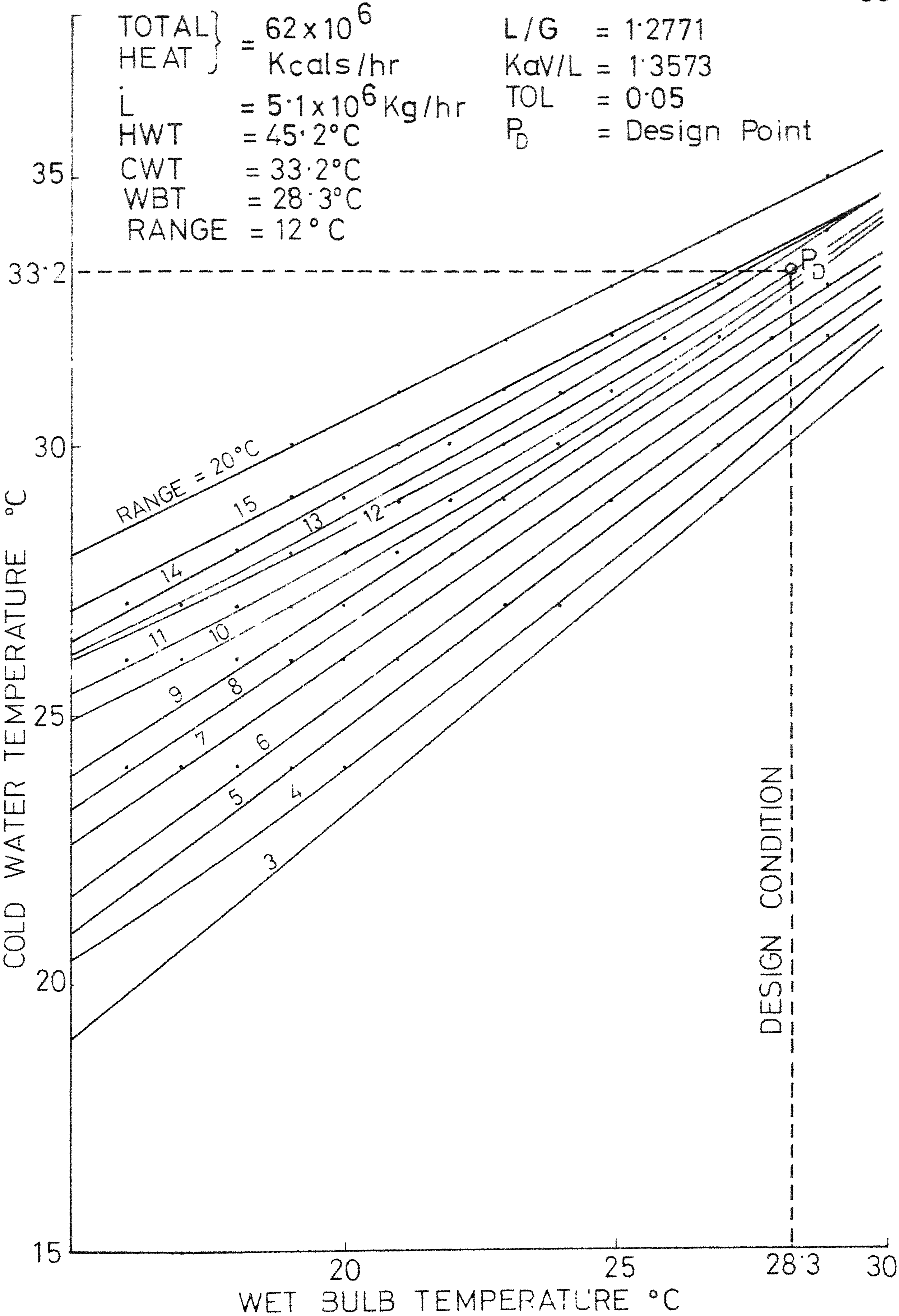


FIG. 3·6 COUNTER FLOW TOWER PERFORMANCE CURVES.
(FERTILIZER PLANT)

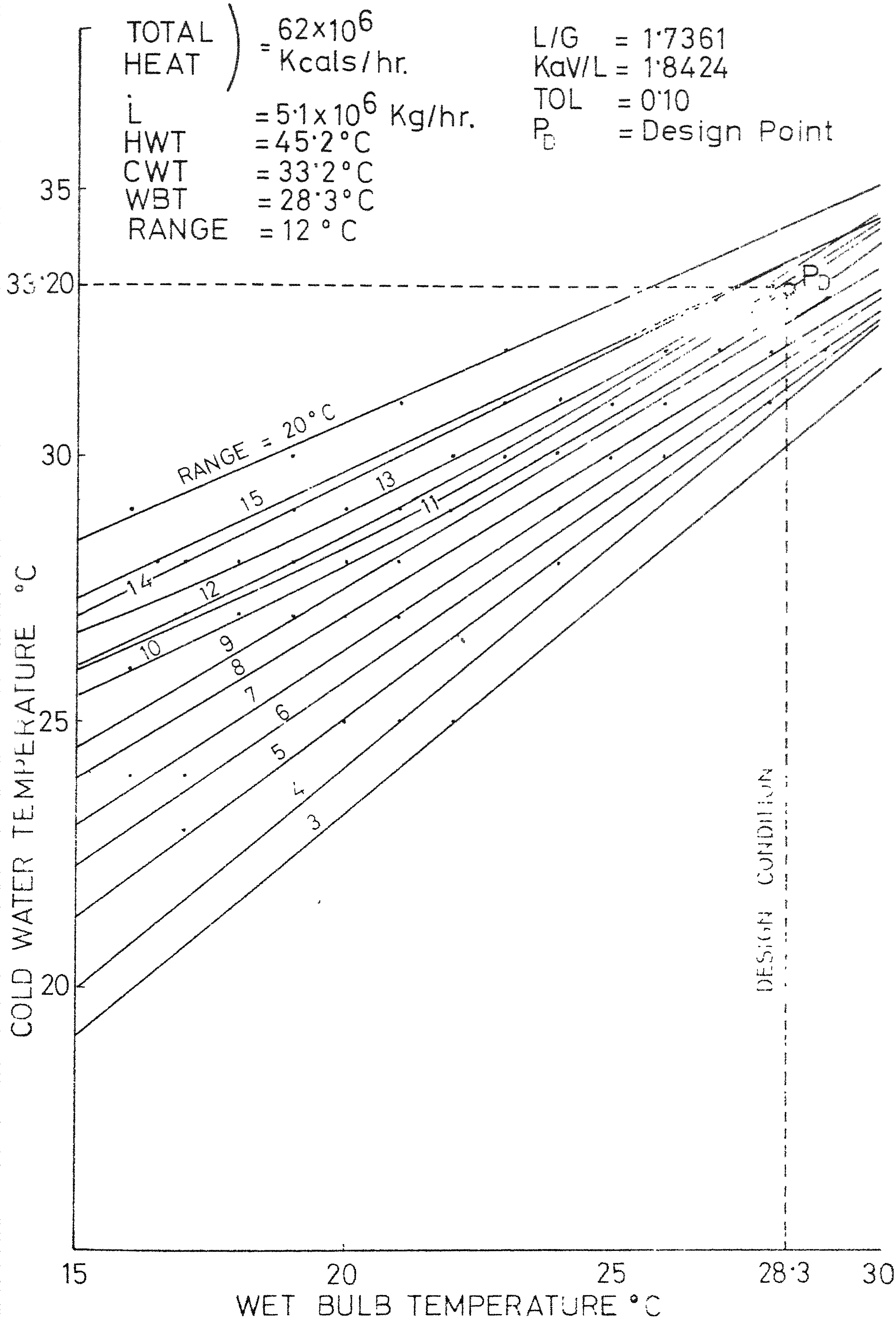


FIG. 3·7 COUNTER FLOW TOWER PERFORMANCE CURVES.
(FERTILIZER PLANT)

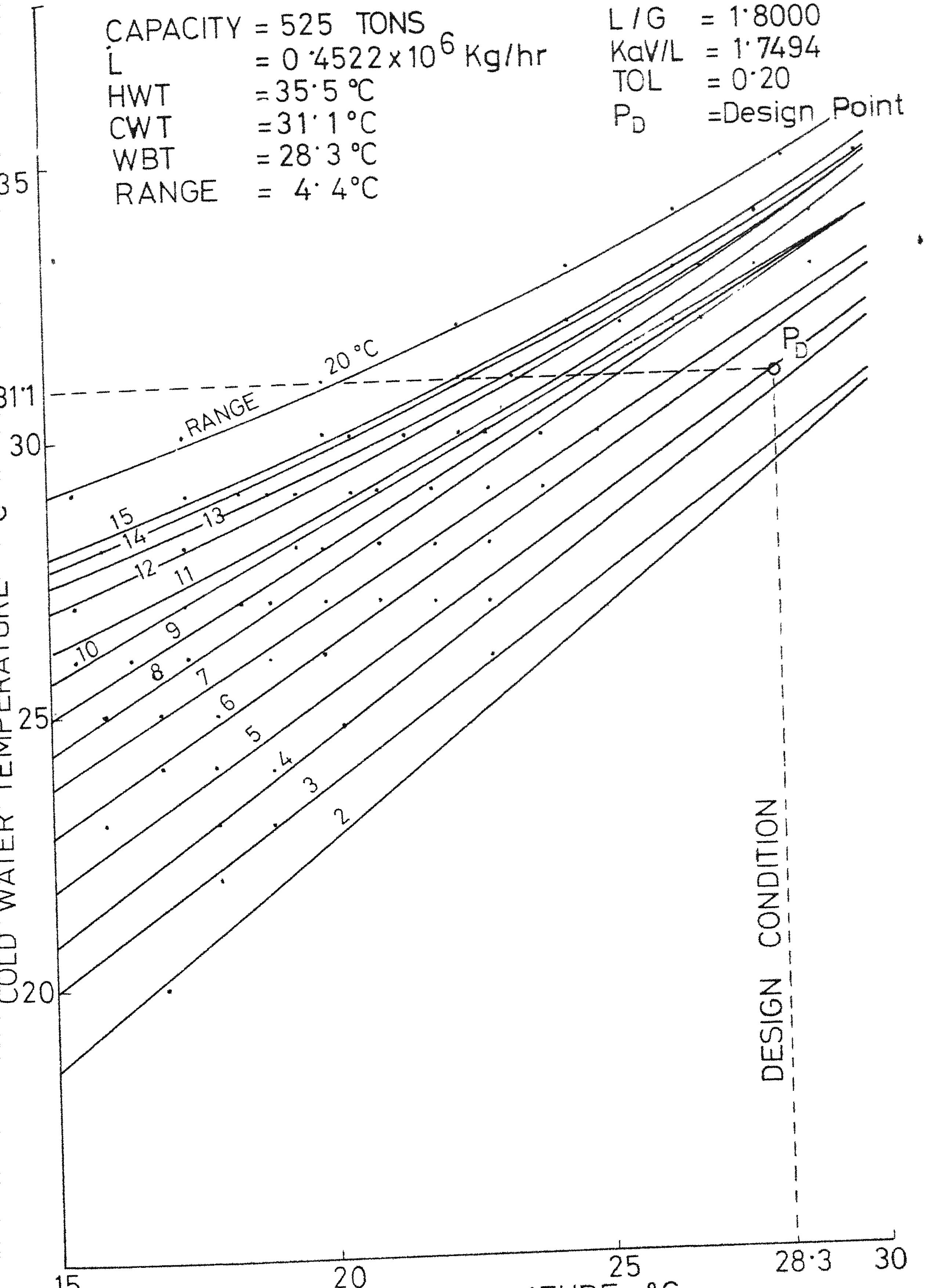


FIG. 3·8 COUNTER FLOW TOWER PERFORMANCE CURVES.
(AIR CONDITIONING PLANT)

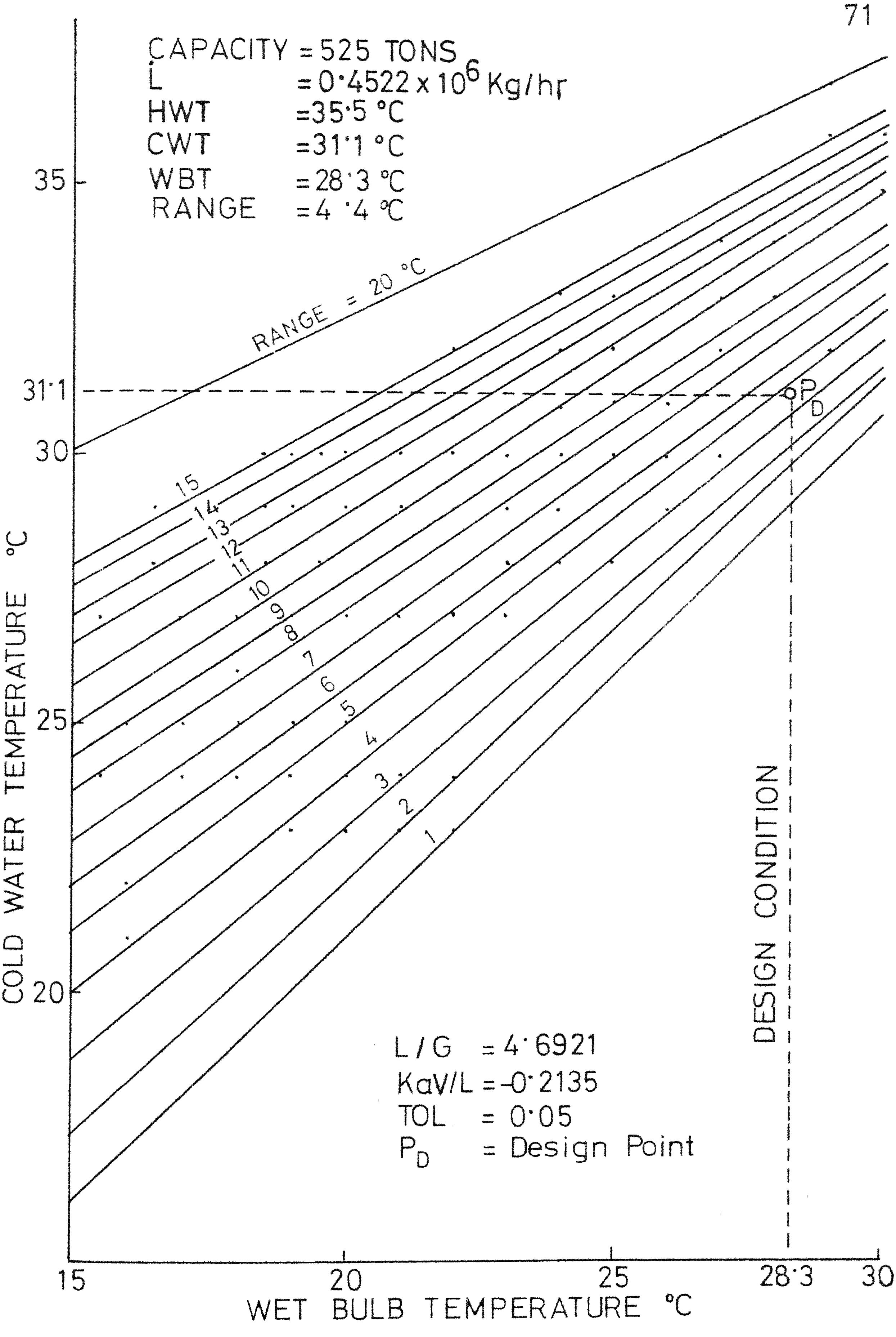


FIG. 3·9 COUNTER FLOW TOWER PERFORMANCE CURVES.
(AIR CONDITIONING PLANT)

(i) Wet bulb temperature: $24.5^{o}C$, $26.2^{o}C$, $27.8^{o}C$ and $28.3^{o}C$.

(ii) Cold water temperature: $15^{o}C$ - $40^{o}C$ by an increment of $1^{o}C$.

(iii) Cooling range: $1^{o}C$ - $20^{o}C$ by an increment of $1^{o}C$.

(iv) L/G ratio: 1.0 - 3.0 by an increment of 0.4.

A computer program has been written (Appendix-B) to evaluate the above values. The time required to compute these values was found to be the same as in the previous case.

In plotting the curves, the L/G ratio is taken as abscissa and $\frac{K\ a\ V}{L}$ as the ordinate. The wet bulb temperature and the cooling range are kept constant in drawing a particular set of performance curves. Figures (3.10) - (3.13) represent sets of performance curves for various cooling ranges and wet bulb temperatures.

CROSS FLOW PERFORMANCE CURVES: As already mentioned in the previous chapter, cross flow tower involves a two-dimensional flow pattern in which water falls downward through the tower and air is drawn horizontally through the packing. The enthalpy of the air changes not only in the vertical direction but also in the horizontal direction.

Consider a vertical section through a cross flow cooling tower as shown in figure (2.4). The width of the tower is taken as unity. The positive X-direction

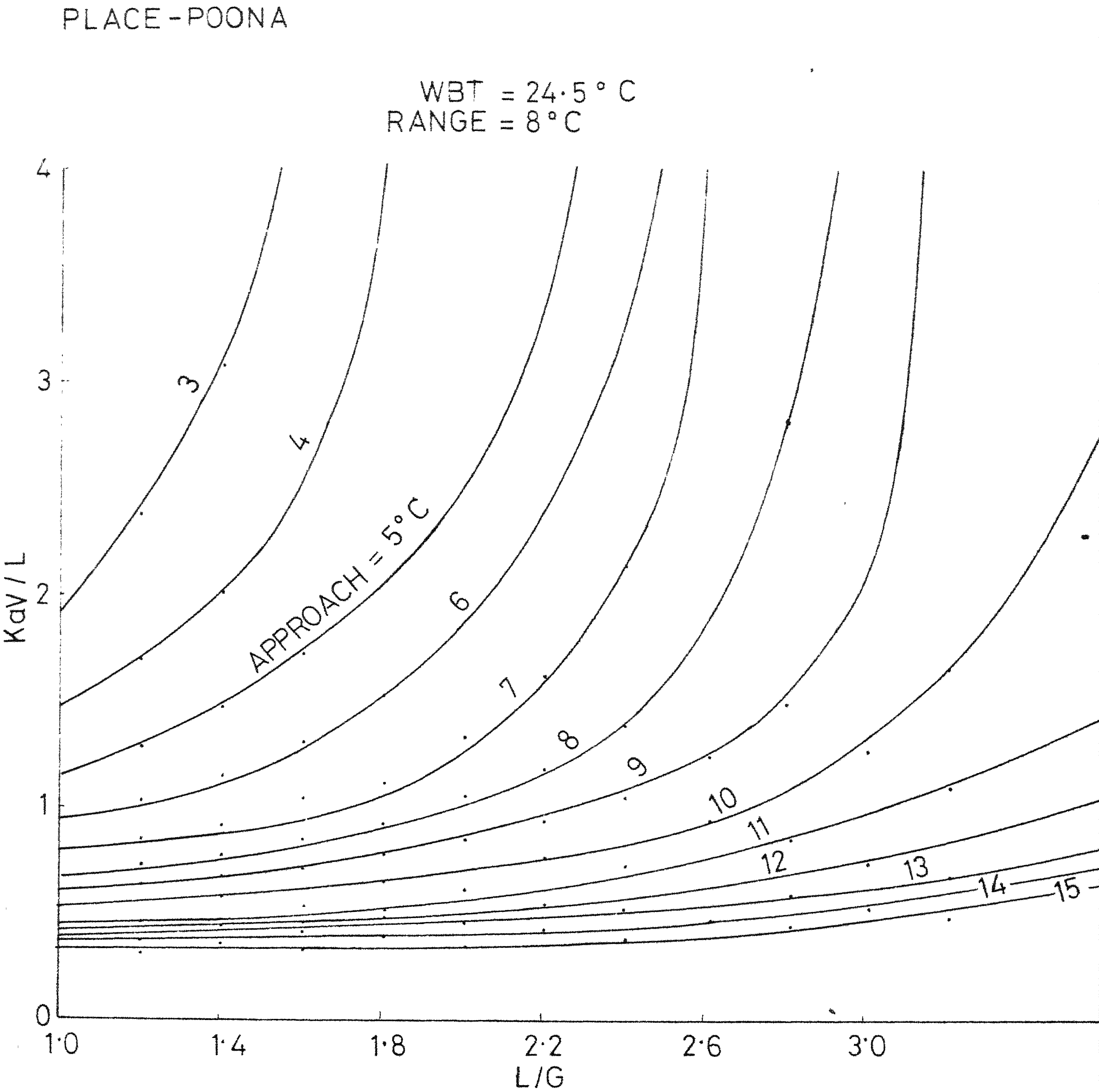


FIG. 3·10 COUNTER FLOW TOWER PERFORMANCE CURVES

PLACES - JAIPUR, NAGPUR

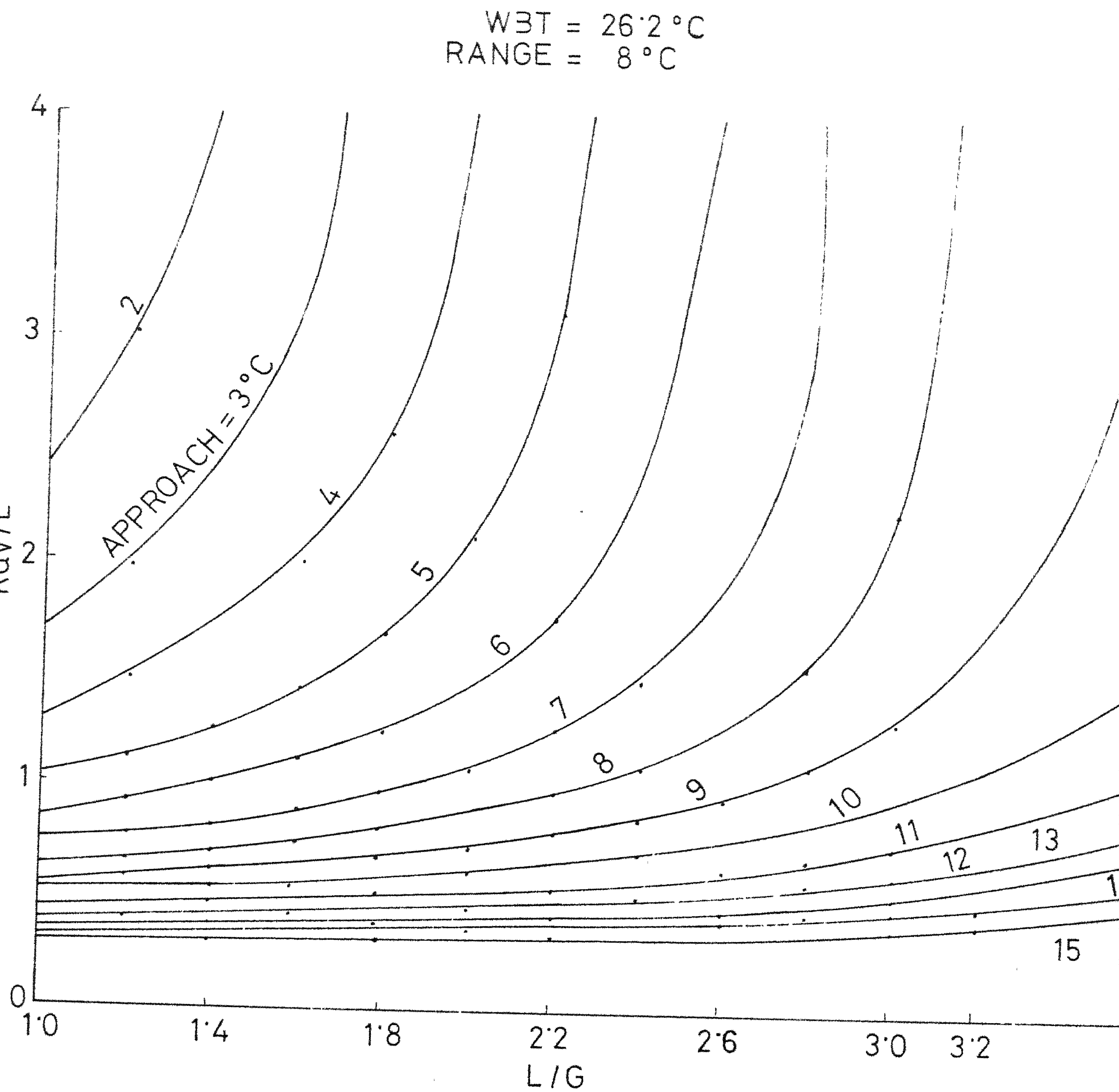


FIG. 3·11 COUNTER FLOW TOWER PERFORMANCE CURVE

PLACES - AHMEDABAD, BOMBAY, GWALIOR, JODHPUR, TRIVENDRUM

WBT = 27·8°C
RANGE = 8°C

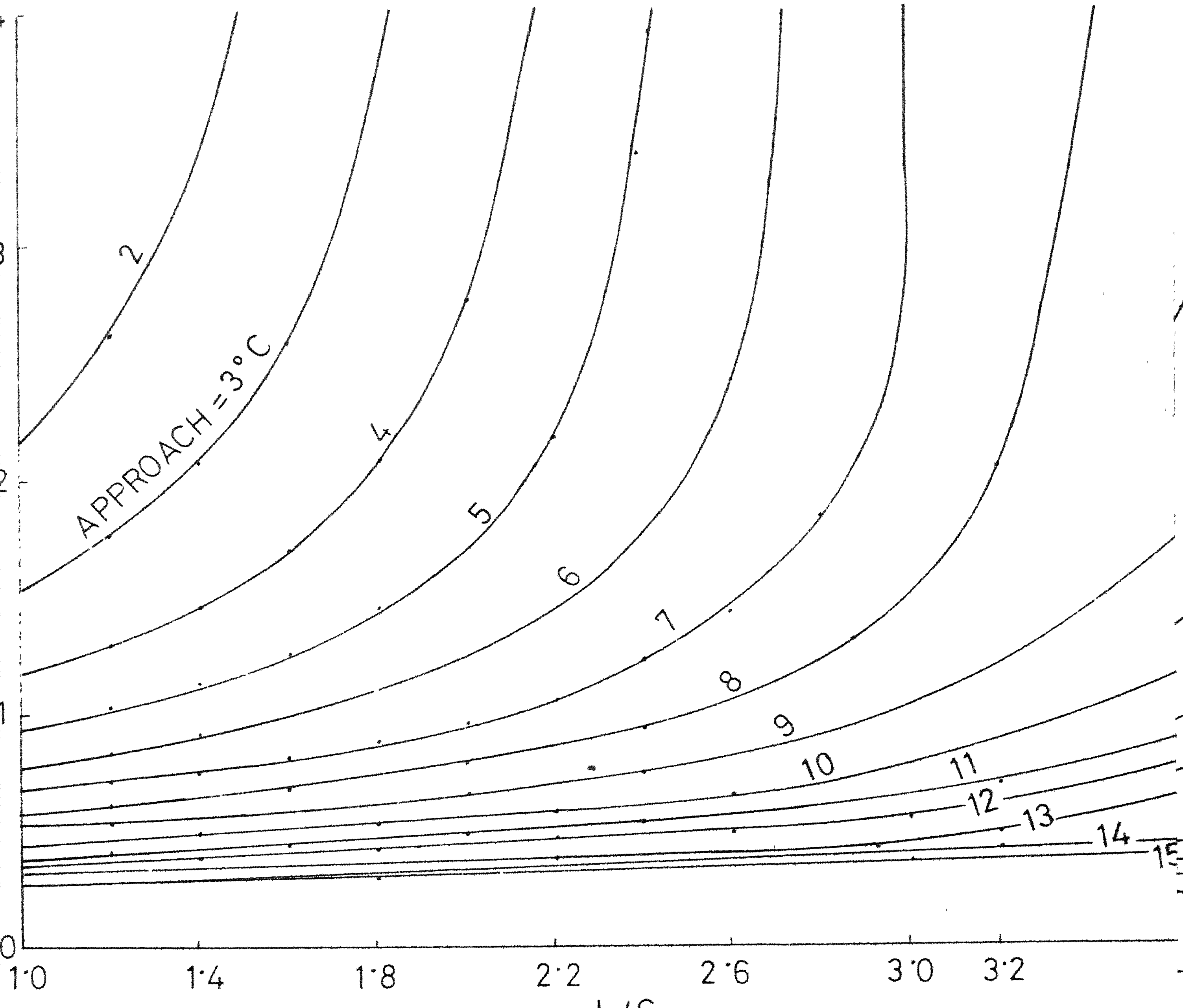


FIG. 3·12 COUNTER FLOW TOWER PERFORMANCE CURVES

PLACES-AMRITSAR, ALLAHABAD, ASANSOL, CALCUTTA, DELHI, GAYA, GAUHATI, JAMSEDPUR, KANPUR, LUCKNOW, MADRAS, PATNA, VISHAKAPATTNAM

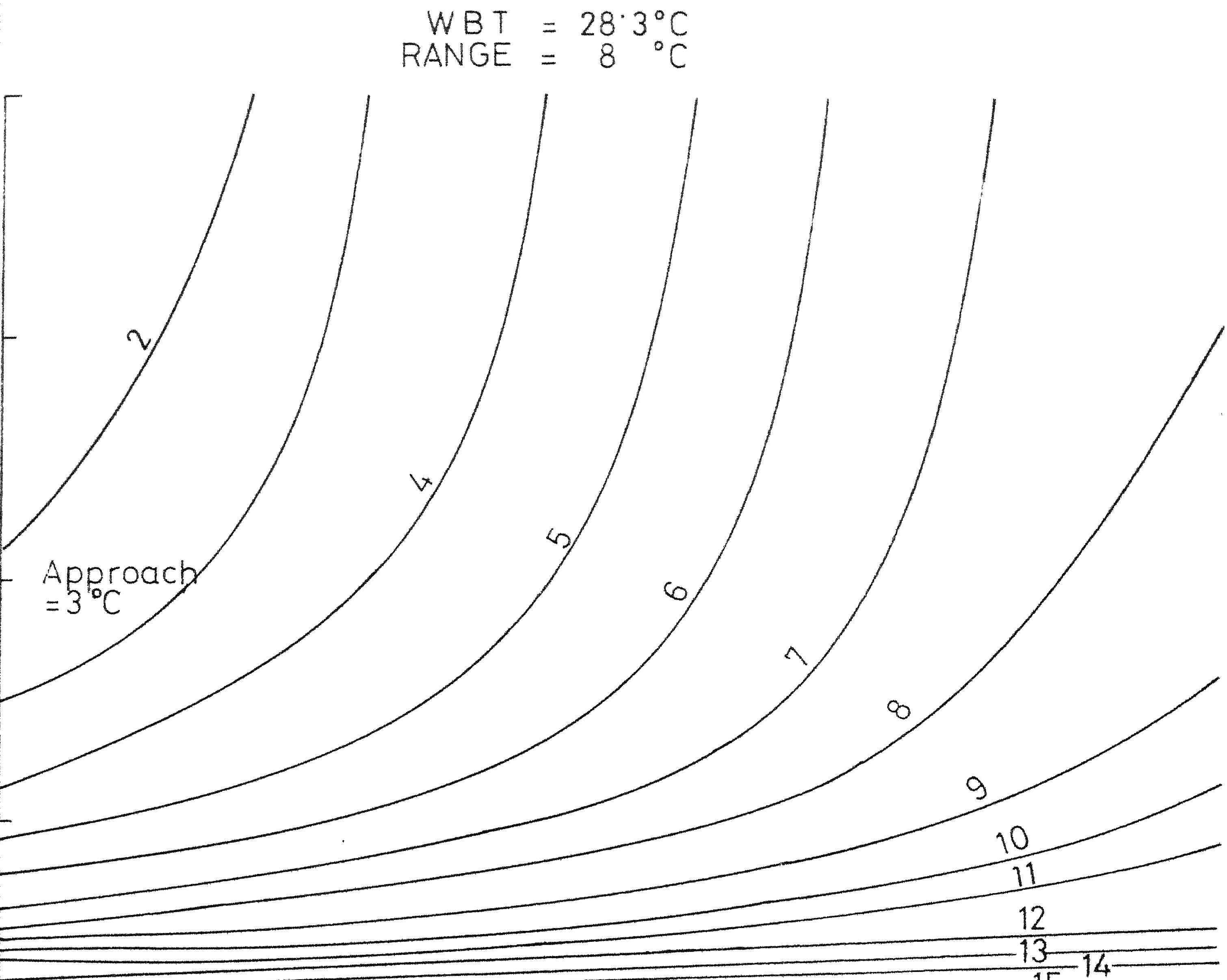


FIG. 3·13 COUNTER FLOW TOWER PERFORMANCE CURVES.

is defined as the direction of the air flow, and the positive Z-direction as the direction of water flow.

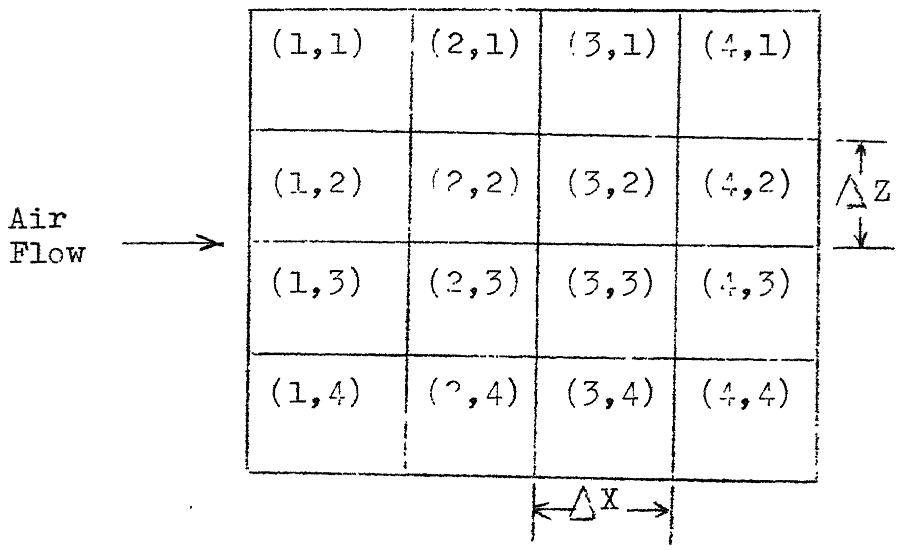


Fig. (2.4), Vertical section through a cross flow cooling tower.

Assumptions:

(1) The hot water temperature entering the top of the tower is at the same temperature along the entire X-boundary in the direction of air flow.

(2) Entering wet bulb temperature of air is constant along the vertical Z-boundary.

Consider a small element of volume of the fill, having dimensions ΔX and ΔZ, respectively, as shown in figure (3.14). The water temperatures entering and leaving the element are T_W and $T_W + \Delta T_W$, respectively. ΔT_W is found to be negative. Air enthalpies entering and leaving the element are H_a

and $H_a + \Delta H_a$ respectively, where ΔH_a is positive.

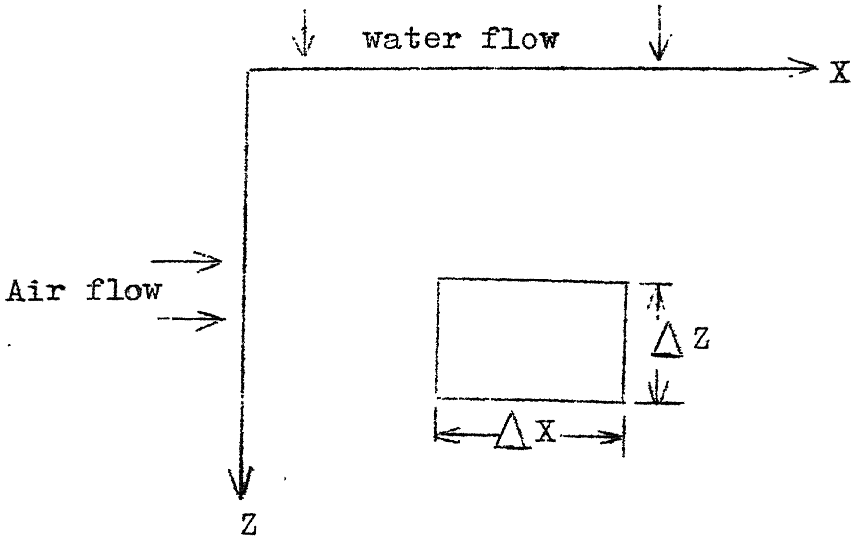


Fig. (3.14), Co-ordinates of a volume element.

Therefore, energy balance for the incremental element is:

$$C_w L \Delta X [T_w - (T_w + \Delta T_w)] = G \Delta Z[(H_a + \Delta H_a) - H_a] \quad (3.13)$$

where,

C_w = Unit heat capacity of water, kcal/kg ^{o}C.

L = Water flow rate, kg/hr.m^2 horizontal area.

T_w = Temperature of water, ^{o}C.

G = Air flow rate, kg/hr.m^2 vertical area.

H_a = Enthalpy of air, kcal/kg dry air.

$$\therefore \quad - L \Delta X \Delta T_w = G \Delta Z \Delta H_a \quad (3.14)$$

or

$$\frac{\Delta T_w}{\Delta Z} = - \frac{G}{L} \frac{\Delta H_a}{\Delta X} \quad (3.15)$$

If the distances $\triangle X$ and $\triangle Z$ are reduced, approaching zero in the limit, eqn. (3.15) becomes

$$\frac{\partial T_w}{\partial Z} = -\frac{G}{L}\frac{\partial H_a}{\partial X} \tag{3.16}$$

Equation (3.16) is the partial differential equation relating the rate of water temperature decrease with the rate of air enthalpy increase.

When the transfer of heat takes place between the unsaturated air and a wetted surface (water surface), the driving force is the "enthalpy potential". It is defined as the difference between the enthalpy of air at the water temperature and that at the actual local wet bulb temperature. Therefore, using enthalpy potential as the driving force for heat transfer, the rate of heat transfer in the volume element is:

$$\triangle q = L\ C_w \triangle X\ (-\triangle T_w) = K_a \triangle X \triangle Z\ (H_w - H_a) \tag{3.17}$$

where,

$\triangle q$ = Heat transfer in the incremental volume $\triangle X \triangle Z$, kcal/hr.*

K_a = Volumetric heat transfer coefficient for a specified cross flow packing, kcal/hr.m^3 (kcal/kg).

---

* Volumetric heat transfer coefficient has been explained in Appendix-D.

H_w = Enthalpy of saturated air at water temperature, kcal/kg dry air.

The value of K_a is determined experimentally. It is a function of water velocity, air velocity, water temperature and packing.

Equation (3.17) can be written as:

$$- L \frac{\Delta T_w}{\Delta Z} = K_a \; (H_w - H_a) \qquad (3.18)$$

If the distance ΔZ is reduced, approaching zero in the limit, equation (3.18) will be

$$\frac{\partial T_w}{\partial Z} = - \frac{K_a}{L} \; (H_w - H_a) \qquad (3.19)$$

Equation (3.19) is the partial differential equation relating the rate of water temperature decrease to the magnitude of the local driving force, $H_w - H_a$.

Combining equations (3.15) and (3.18), we have

$$G \; \frac{\Delta H_a}{\Delta X} = K_a \; (H_w - H_a) \qquad (3.20)$$

$$\Delta H_a = \frac{K_a \, \Delta X}{G} \; (H_w - H_a)$$

$$= \Delta \bar{X} \; (H_w - H_a) \qquad (3.21)$$

where,

$\Delta \bar{X} = \frac{K_a \, \Delta X}{G}$, represents a non-dimension-distance.

If we consider two points ''1'' and ''2'' separated by the non-dimensional distance $\Delta \bar{X}$ along the X-

boundary of the tower, the eqn. (3.21) may be expressed as:

$$\triangle H_a = \triangle \bar{X} \left[\frac{H_{w1} + H_{w2}}{2} - \frac{H_{a1} + H_{a2}}{2} \right]$$

$$= \frac{\triangle \bar{X}}{2} \left[(H_w - H_a)_1 - (H_w - H_a)_2 \right] \qquad (3.22)$$

Thus, the enthalpy difference for air is equal to the product of the average of the enthalpy potentials at the two successive horizontal points and the dimensionless distance between them.

In computing the air enthalpy for the points on the X - boundary, it is assumed that the hot water temperature entering the top of the tower is the same along the entire length in the direction of air flow.

For a generalised solution of the air enthalpy along the X-boundary of the tower, consider the points along the X and Z boundaries as shown in figure (3.15)

Applying the equation (3.22) for the points (i,j) and (i-1,j) in figure (3.15):

$$H_{a(i,j)} - H_{a(i-1,j)} = \frac{\triangle \bar{X}}{2} \left[(H_w - H_a)_{(i-1,j)} + (H_w - H_a)_{(i,j)} \right] \qquad (3.23)$$

$$= \frac{\triangle \bar{X}}{2} \left[H_{w(i-1,j)} + H_{w(i,j)} - H_{a(i-1,j)} - H_{a(i,j)} \right] \qquad (3.24)$$

$$\therefore\ H_{a(i,j)} + \frac{\Delta \bar{X}}{2} H_{a(i,j)} = H_{a(i-1,j)} + \frac{\Delta \bar{X}}{2} \Big[H_{w(i-1,j)} + H_{w(i-j)} - H_{a(i-1,j)} \Big]$$

$$\therefore\ H_{a(i,j)} = \frac{H_{a(i-1,j)} + \frac{\Delta \bar{X}}{2} \Big[H_{w(i-1,j)} + H_{w(i,j)} - H_{a(i-1,j)} \Big]}{\Big[1 + \frac{\Delta \bar{X}}{2} \Big]} \qquad (3.25)$$

i-1, j-1	i, j-1	i+1, j-1	i+2, j-1	i+3, j-1	
i-1, j	i, j	i+1, j	i+2, j	i+3, j	
i-1, j+1	i, j+1	i+1, j+1	i+2, j+1	i+3, j+1	
i-1, j+2	i, j+2	i+1, j+2	i+2, j+2	i+3, j+2	
i-1, j+3	i, j+3	i+1, j+3	i+2, j+3	i+3, j+3	

Fig.(3.15), Two-dimensional arrangement of points in a cross flow tower.

The equation (3.25) is used to determine the air enthalpy along the X-boundary. Since, it is assumed that the hot water temperature remains constant in the X-direction, the calculations are started with the following values of i and j

$$i = 1.0, \quad j = 1.0, \quad \Delta i = 1.0, \quad \Delta j = 0.0$$

Similarly, the water temperature is calculated for the points on Z - boundary knowing that the entering wet bulb temperature is constant along Z - boundary. Considering equation (3.18),

$$- L \frac{\Delta T_W}{\Delta Z} = K_a (H_W - H_a)$$

$$\Delta T_W = - \frac{K_a \Delta Z}{L} (H_W - H_a) \qquad (3.26)$$

$$= - \Delta \bar{Z} (H_W - H_a) \qquad (3.27)^{*}$$

where, $\Delta \bar{Z} = \frac{K_a \Delta Z}{L}$

In equation (3.27), $\Delta \bar{Z}$ is a non-dimensional tower length. Again, considering the points ''1'' and ''2'' in the vertical Z-direction, separated by a dimensionless distance $\Delta \bar{Z}$, the expression (3.27) may be written as:

$$\Delta T_W = - \Delta \bar{Z} \left[\frac{H_{W1} + H_{W2}}{2} - \frac{H_{a1} + H_{a2}}{2} \right]$$

$$= - \frac{\Delta \bar{Z}}{2} \left[(H_W - H_a)_1 - (H_W - H_a)_2 \right] \qquad (3.28)$$

Thus, the enthalpy difference for water is equal to the product of the average of the enthalpy potentials at two successive vertical points and the dimensionless distance $\Delta \bar{Z}$ between those two successive points. Referring figure (3.15), and the points (i,j-1) and (i,j), the equation (3.28) becomes:

---

* In deriving this relation the L.H.S. has been nullified by the specific heat of water C_W which is unity.

$$T_{w(i,j)} - T_{w(i,j-1)} =$$

$$\frac{\Delta \bar{Z}}{2} \cdot \left[(H_w - H_a)_{(i,j-1)} + (H_w - H_a)_{(i,j)} \right] \tag{3.29}$$

$$\therefore \; T_{w(i,j)} = T_{w(i,j-1)} - \frac{\Delta \bar{Z}}{2} \left[H_{w(i,j-1)} + H_{w(i,j)} - H_{a(i,j-1)} - H_{a(i,j)} \right] \tag{3.30}$$

Since, it is assumed that the entering wet bulb temperature is constant in the Z direction, the calculations are started with the following values of i and j

$$i = 1.0, \quad j = 1.0, \quad \Delta i = 0.0, \quad \Delta j = 1.0$$

To solve equations (3.25) and (3.30), a computer program has been developed using IBM 7044/1401 computers. The procedure to calculate $H_{a(i,j)}$ and $T_{w(i,j)}$ is as follows:

<u>X boundary</u>

The program begins by reading the initial hot water temperature entering the top of the tower and the initial wet bulb temperature of air. These are denoted by $T_{w(i,j-1)}$ and $H_{a(i-1,j)}$, respectively. The program also reads the enthalpy of air at the entering wet bulb temperature. $H_{a(i,j)}$ is solved by using the equation (3.25), and having an increment $\Delta i = 1.0$ every time.

Z-boundary

$T_{w(i,j)}$ is calculated by using equation (3.30) as follows:

(i) Assume $T_{w(i,j)} = T_{wb}$ (wet bulb temperature of the entering air, ^{o}C).

(ii) For the assumed value of $T_{w(i,j)}$, $H_{w(i,j)}$ is calculated by using the standard tables and interpolating results on the computer (Appendix-C_1, subroutine ALPHA).

(iii) $T_{w(i,j)}$ is computed by substituting $H_{w(i,j)}$, in equation (3.30).

(iv) If $T_{w(i,j)\,assumed} = T_{w(i,j)\,calculated}$, within tolerance, we stop and proceed for the next step to calculate $T_{w(i,j)}$ by giving an increment $\Delta j = 1.0$. A tolerance of 0.01 is assumed for accuracy.

(v) If the tolerance is not met,

$$\text{assume } T_{w(i,j)} = \frac{T_{w(i,j)\,assumed} + T_{w(i,j)\,calculated}}{2}$$

and the steps (ii), (iii), (iv) and (v) are repeated until the tolerance is met.

In equations (3.25) and (3.30), a value of 0.06 is taken for the dimensionless mesh size $\Delta\overline{X}$ and $\Delta\overline{Z}$.

Interior Points

By taking the known values of T_w, H_w and H_a from the preceding points, the values $H_{a(i,j)}$ and $T_{w(i,j)}$ for

interior points also can be obtained, again by using equations (3.25) and (3.30). Starting with i = 2.0, j = 2.0, both $\triangle i$ and $\triangle j$ are given an increment value of 1.0, simultaneously, and the following steps are adopted:

(i) Assume $[H_{w(i,j)} - H_{a(i,j)}] = [H_{w(i-1,j)} - H_{a(i-1,j)}]$

$\therefore$ equation (3.25) becomes,

$$H_{a(i,j)} = H_{a(i-1,j)} + \triangle \bar{X} \, [H_{w(i-1,j)} - H_{a(i-1,j)}] \qquad (3.31)$$

and, equation (3.30) becomes,

$$T_{w(i,j)} = T_{w(i,j-1)} - \frac{\triangle \bar{Z}}{2} [H_{w(i,j-1)} - H_{a(i,j-1)} + H_{w(i-1,j)} - H_{a(i-1,j)}] \qquad (3.32)$$

(ii) Compute $H_{a(i,j)}$ and $T_{w(i,j)}$ by using equations (3.31) and (3.32)

(iii) For computed value of $T_{w(i,j)}$, $H_{w(i,j)}$ is calculated by using the standard tables and interpolating results on the computer (Appendix-C_1 and Appendix-C_2).

(iv) Calculate $[H_{w(i,j)} - H_{a(i,j)}]$

(v) If $[H_{w(i,j)} - H_{a(i,j)}]_{\text{assumed}} = [H_{w(i,j)} - H_{a(i,j)}]_{\text{calculated}}$, within the tolerance limit, we stop and proceed to the next step of calculation by giving increments for $\triangle i = 1.0$ and $\triangle j = 1.0$. A tolerance

of 0.01 is assumed for accuracy.

(vi) If the tolerance is not met, take the average of the assumed and calculated values of $[H_{w(i,j)} - H_{a(i,j)}]$, and the steps (iii), (iv), (v) and (vi) are repeated until tolerance is met.

For computing the values of H_a, H_w and T_w, a 30 x 30 matrix of points has been considered with a mesh size of 0.06. The elements of the above matrix are divided into groups in a manner such that each element in a certain group gives the same value of the parameter to be determined. This classification is done to simplify the plotting of the performance curves. The computer program developed is given in Appendix - C_1.

The values of unknown parameters are obtained similarly, for different inlet hot water temperatures and different entering wet bulb temperatures.

These points are plotted to obtain a set of curves showing average hot water temperatures and enthalpies of moist air. These curves are dimensionless and are not related to a particular tower design. This permits the results to be applied for any cross flow tower for which K_a, G and L are known.

Figures (3.16) - (3.27) show sets of performance curves for hot water temperatures of 43°C, 35.5°C and 45.2°C and inlet design wet bulb temperatures of 24.5°C,

PLACE- POONA

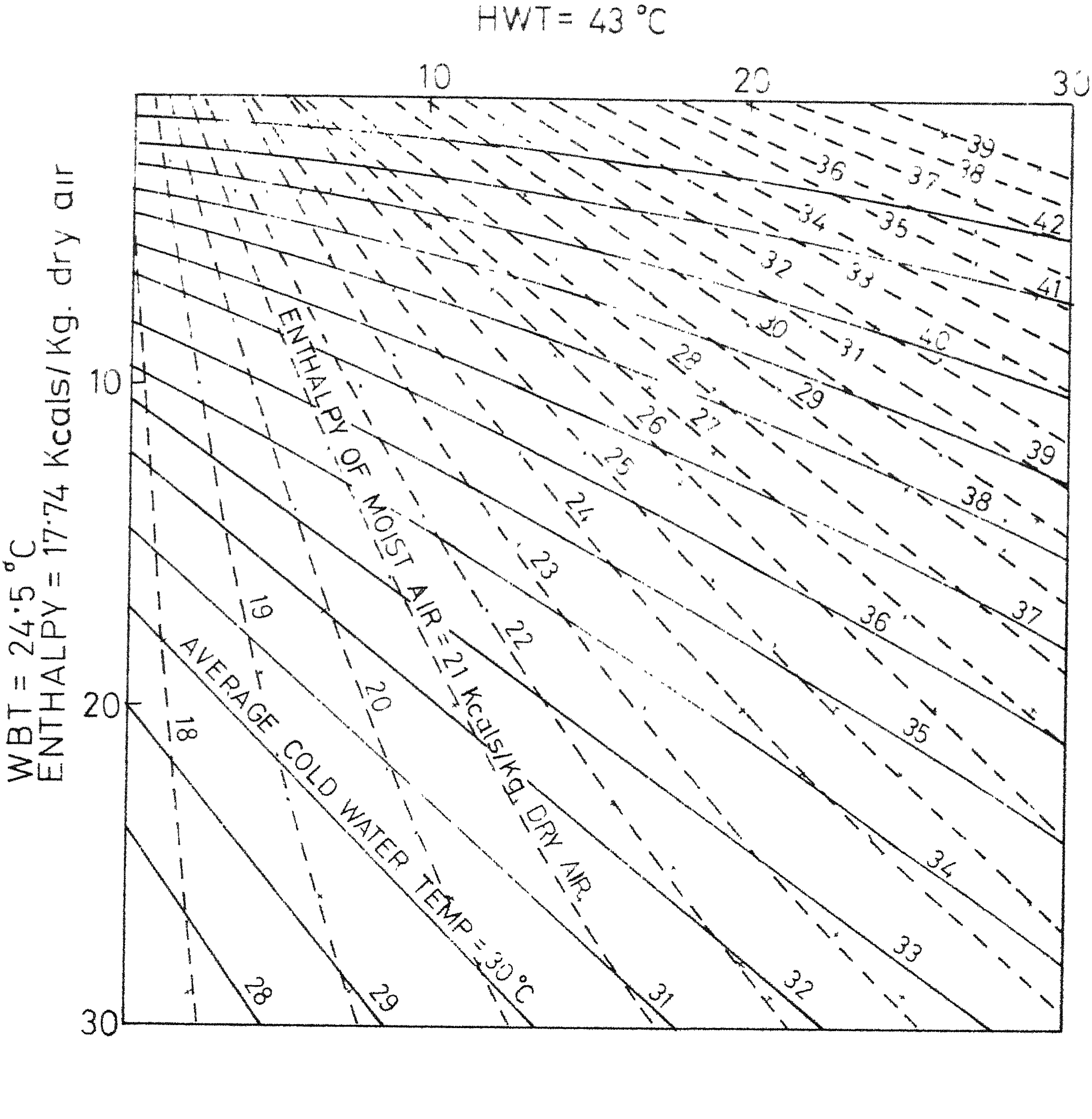


FIG. 3·16 CROSS FLOW TOWER PERFORMANCE CURVES.

(THERMAL POWER PLANT)

PLACES- JAIPUR, NAGPUR

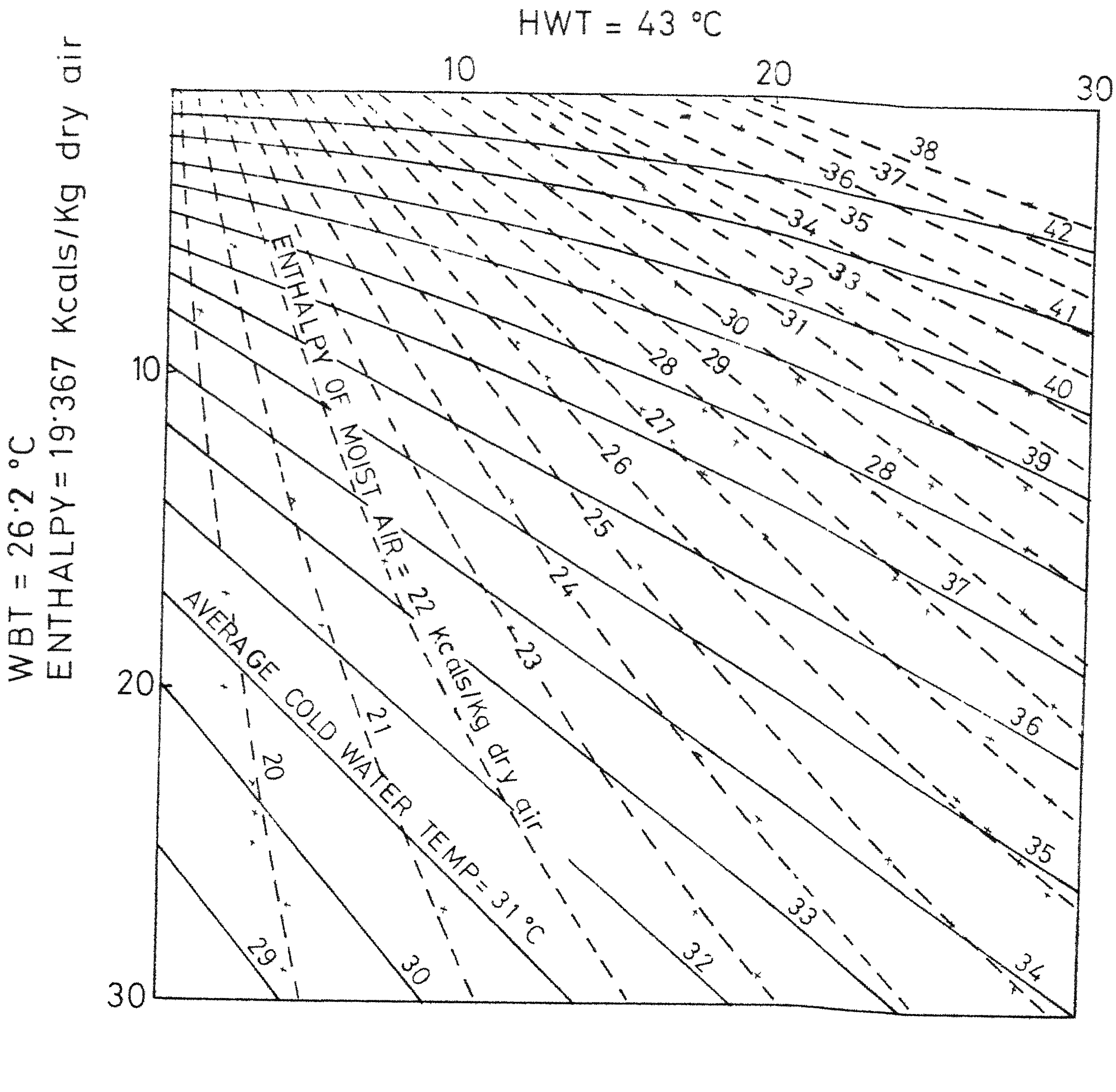


FIG. 3.17 CROSS FLOW TOWER PERFORMANCE CURVES.
(THERMAL POWER PLANT)

PLACES- AHMEDABAD, BOMBAY, GWALIOR, JODHPUR TRIVENDRUM.

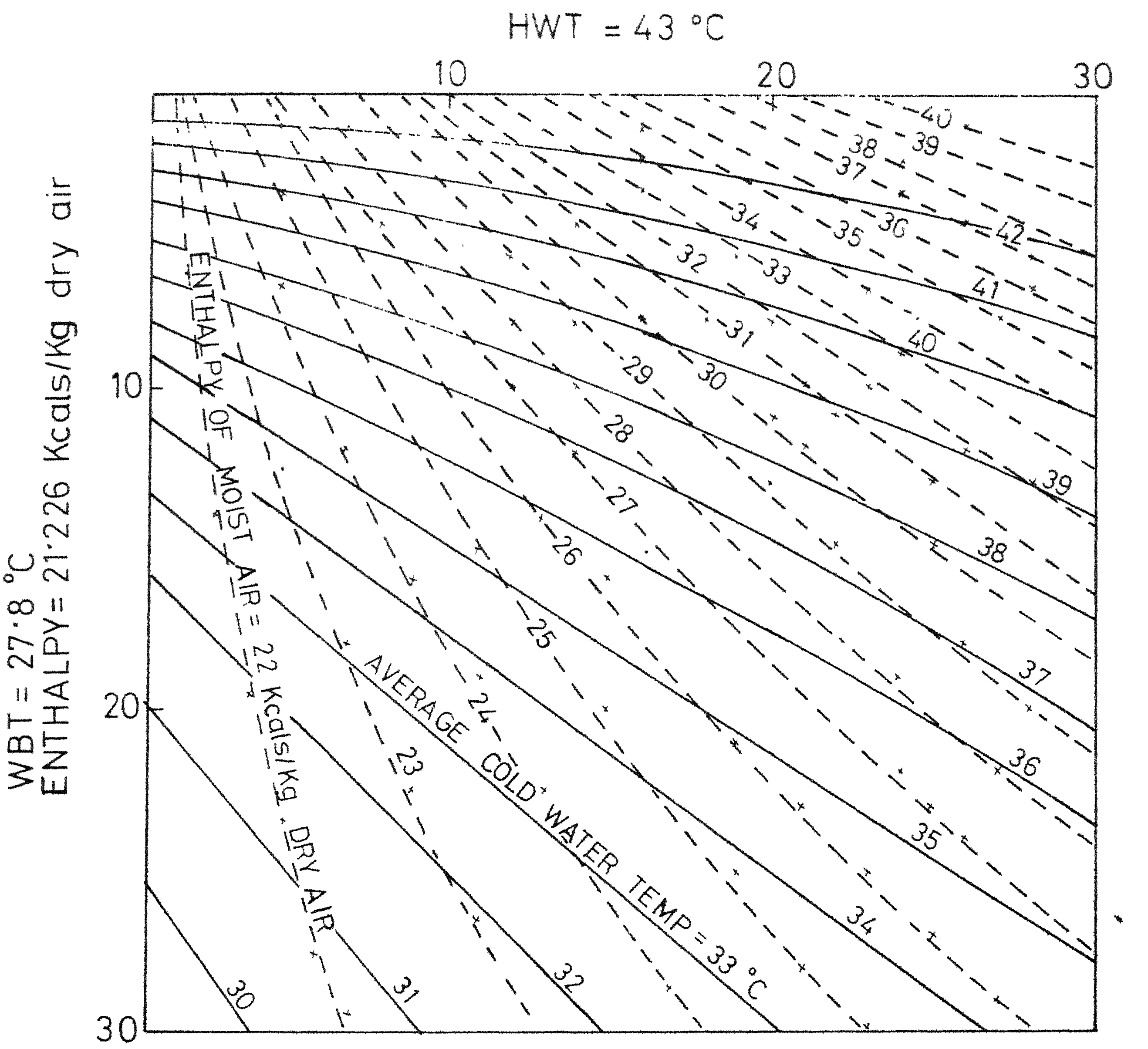


FIG. 3·18 CROSS FLOW TOWER PERFORMANCE CURVES.

(THERMAL POWER PLANT)

PLACES— AMRITSAR, ALLAHABAD, ASANSOL, CALCUTTA, DELHI, GAYA, GAUHATI, JAMSHEDPUR, KANPUR, LUCKNOW, MADRAS, PATNA, VISHAKAPATNAM

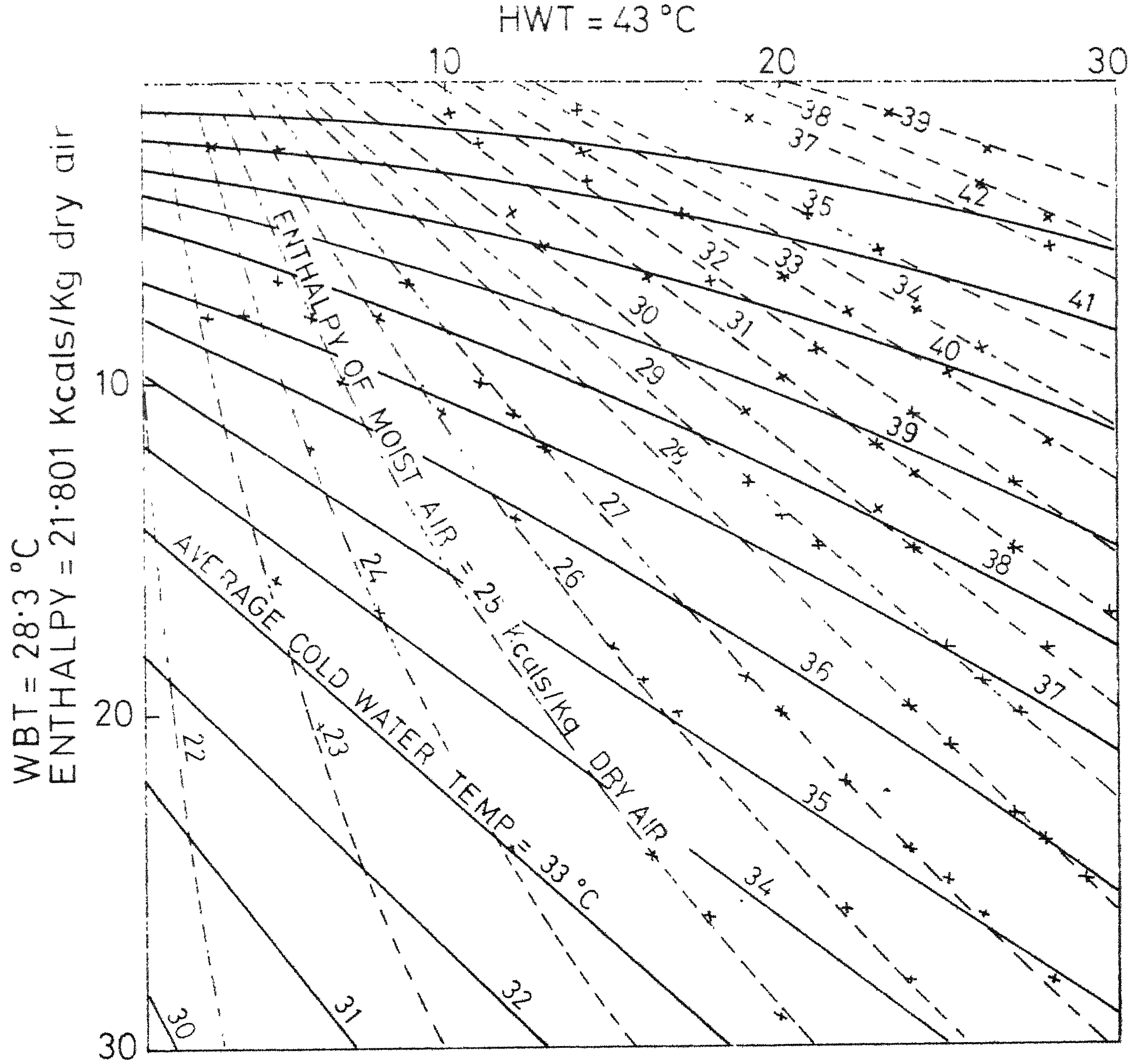


FIG. 3·19 CROSS FLOW TOWER PERFORMANCE CURVES.

PLACE - POONA

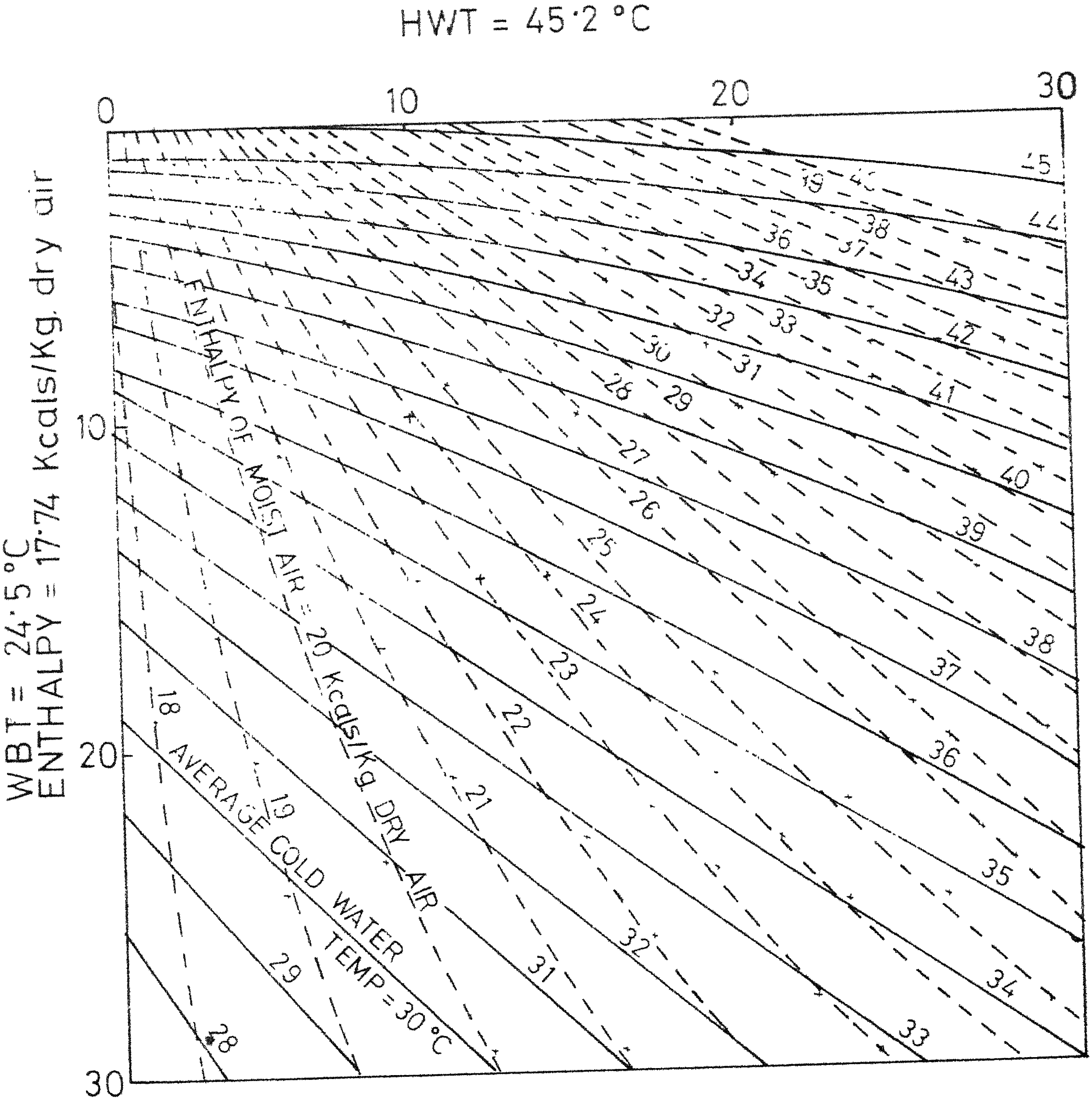


FIG. 3·20 CROSS FLOW COOLING TOWER PERFORMANCE CURVES.

(FERTILIZER PLANT)

PLACES-JAIPUR, NAGPUR

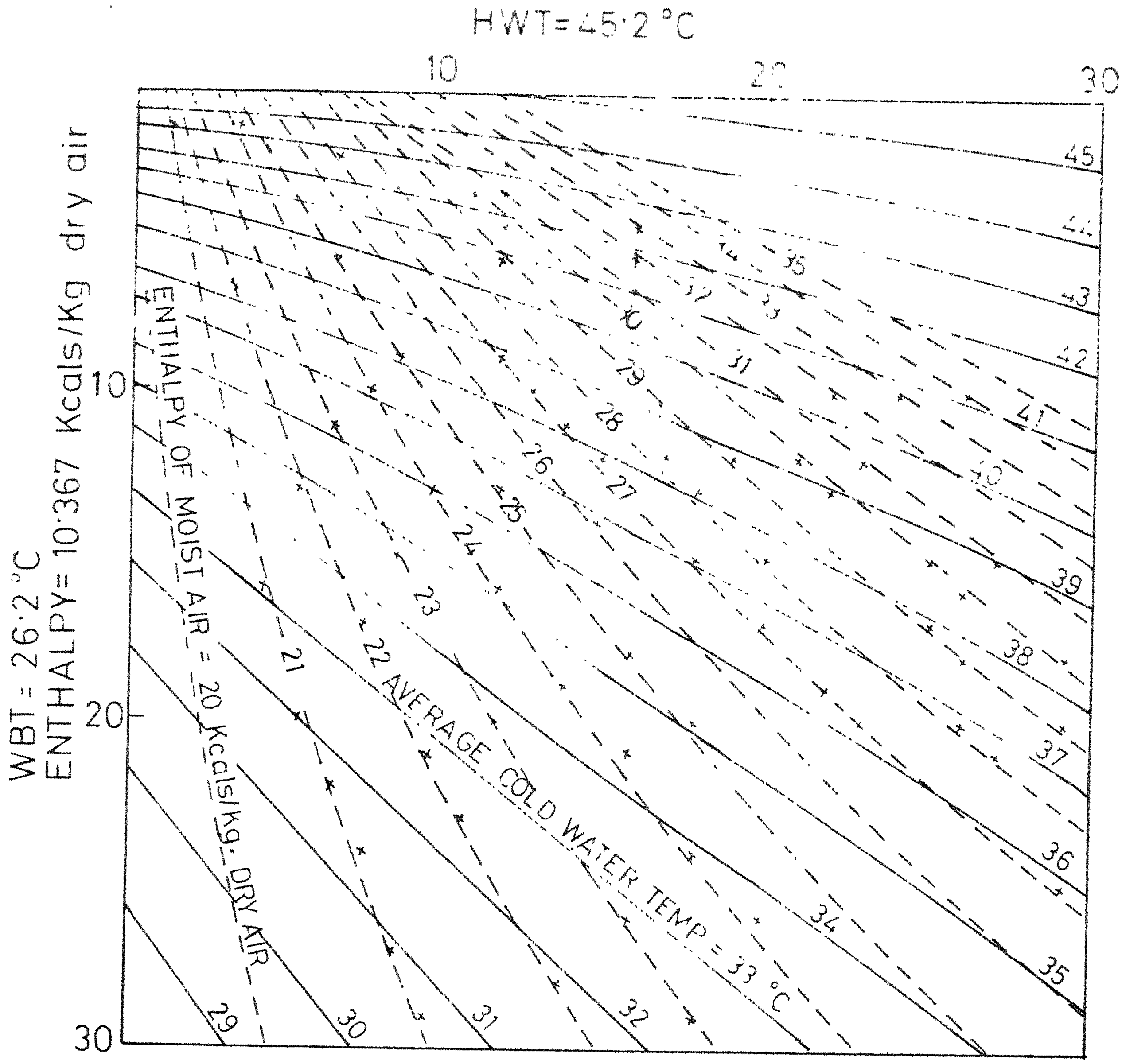


FIG. 3·21 CROSS FLOW COOLING TOWER PERFORMANCE CURVES.

(FERTILIZER PLANT)

PLACES – AHMEDABAD, BOMBAY, GWALIOR, JODHPUR, TRIVENDRUM

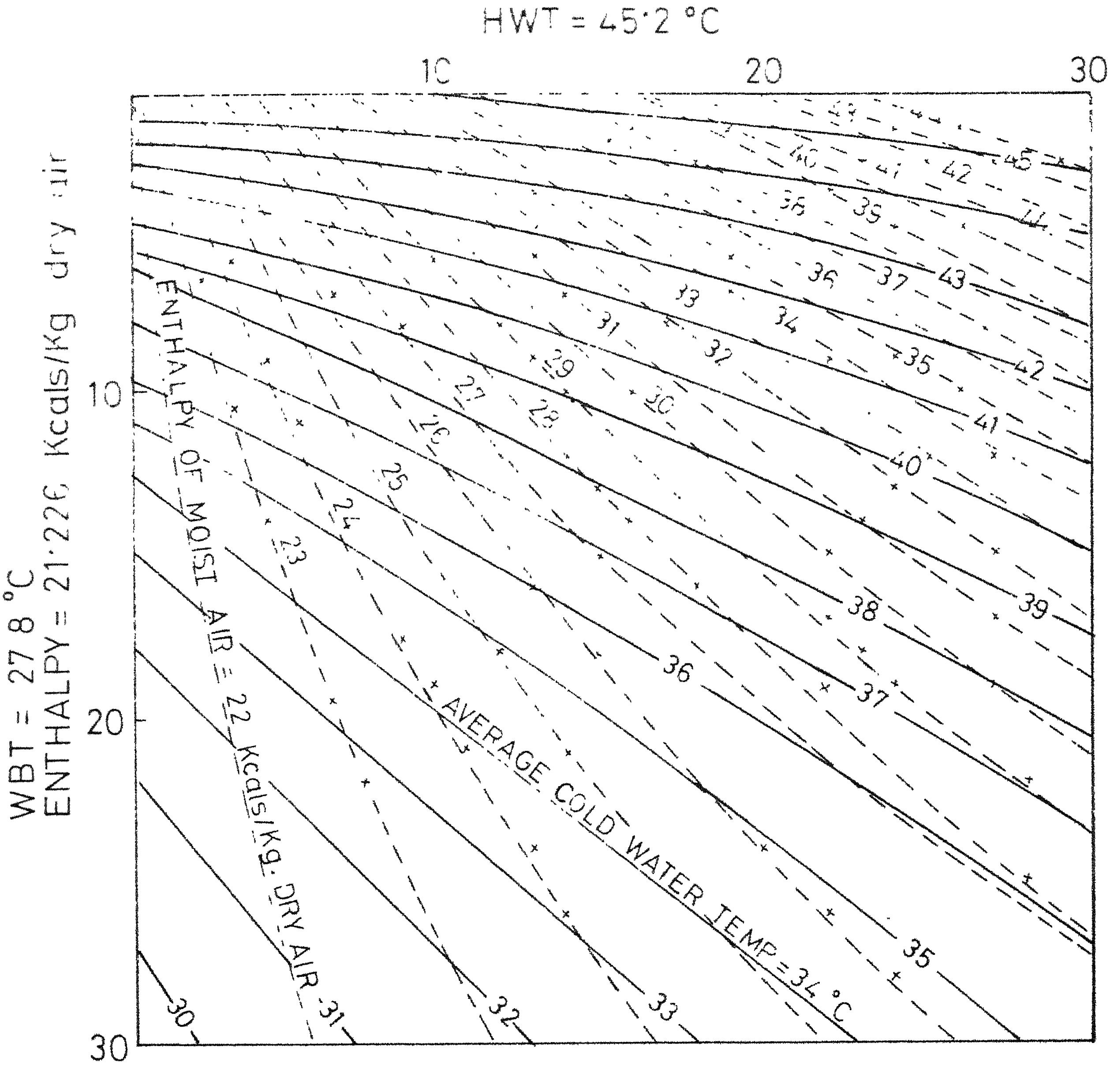


FIG. 3·22 CROSS FLOW TOWER PERFORMANCE CURVES.
(FERTILIZER PLANT)

PLACES- AMRITSAR, ALLAHABAD, ASANSOL, CALCUTTA
DELHI, GAYA, GAUHATI, JAMSHEDPUR, KANPUR
LUCKNOW, MADRAS, PATNA, VISHAKAPATTANAM.

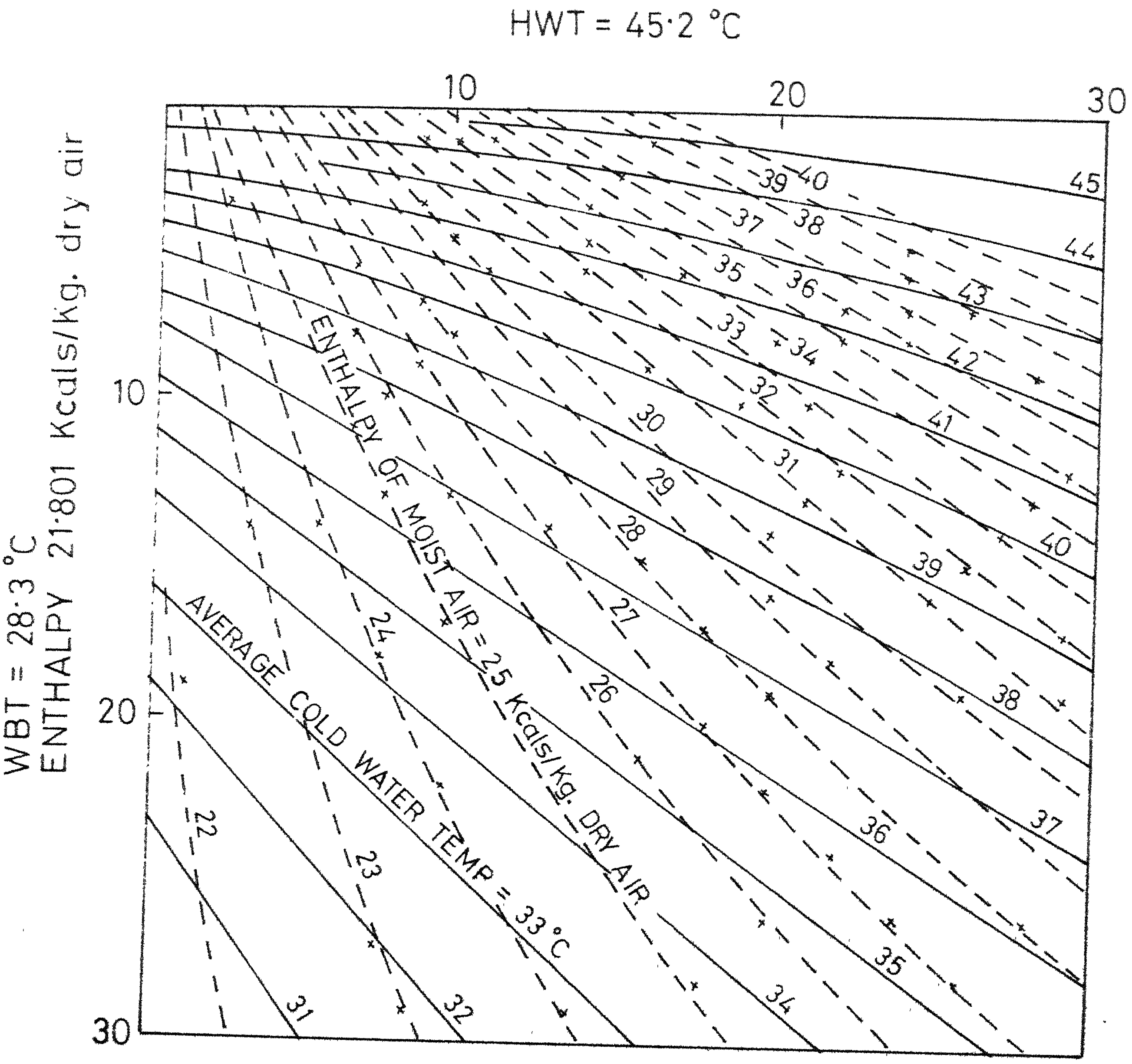


FIG 3·23 CROSS FLOW TOWER PERFORMANCE CURVES.
(FERTILIZER PLANT)

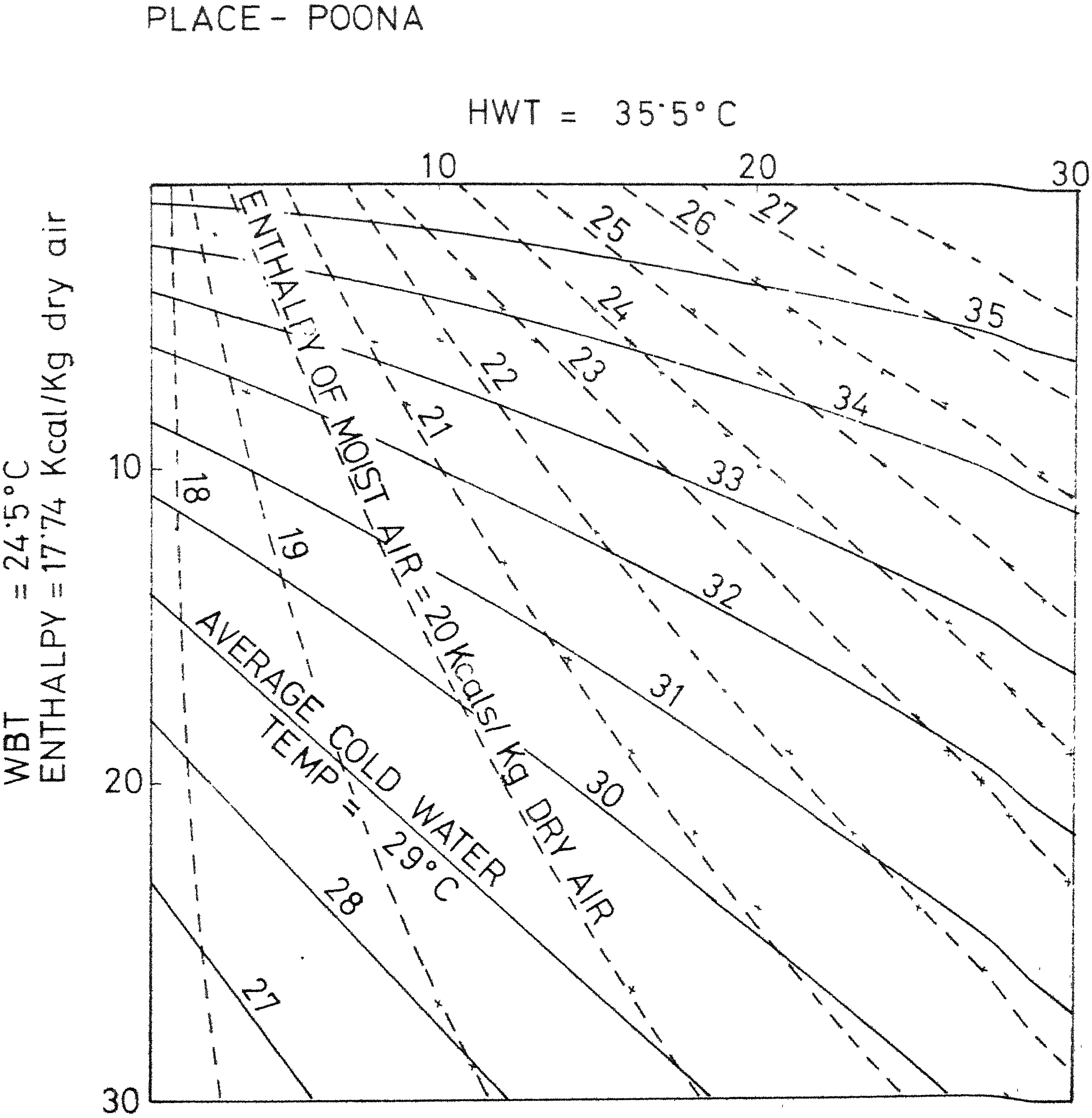


FIG. 3·24 CROSS FLOW TOWER PERFORMANCE CURVES.
(AIR CONDITIONING PLANT)

PLACES - JAIPUR, NAGPUR

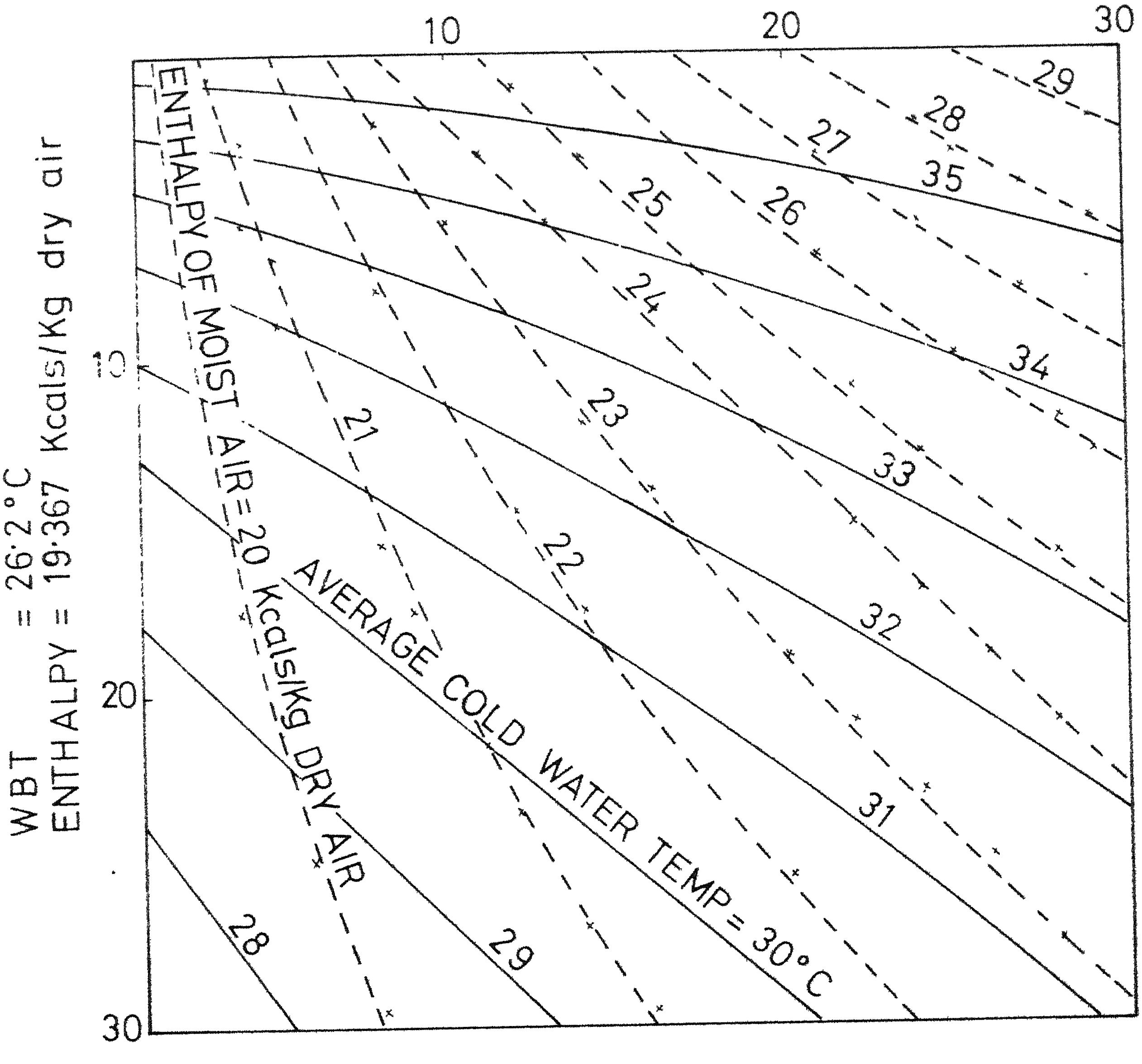


FIG. 3·25 CROSS FLOW TOWER PERFORMANCE CURVES.
(AIR CONDITIONING PLANT)

PLACES - AHMEDABAD, BOMBAY, GWALIOR, JODHPUR, TRIVENDRUM

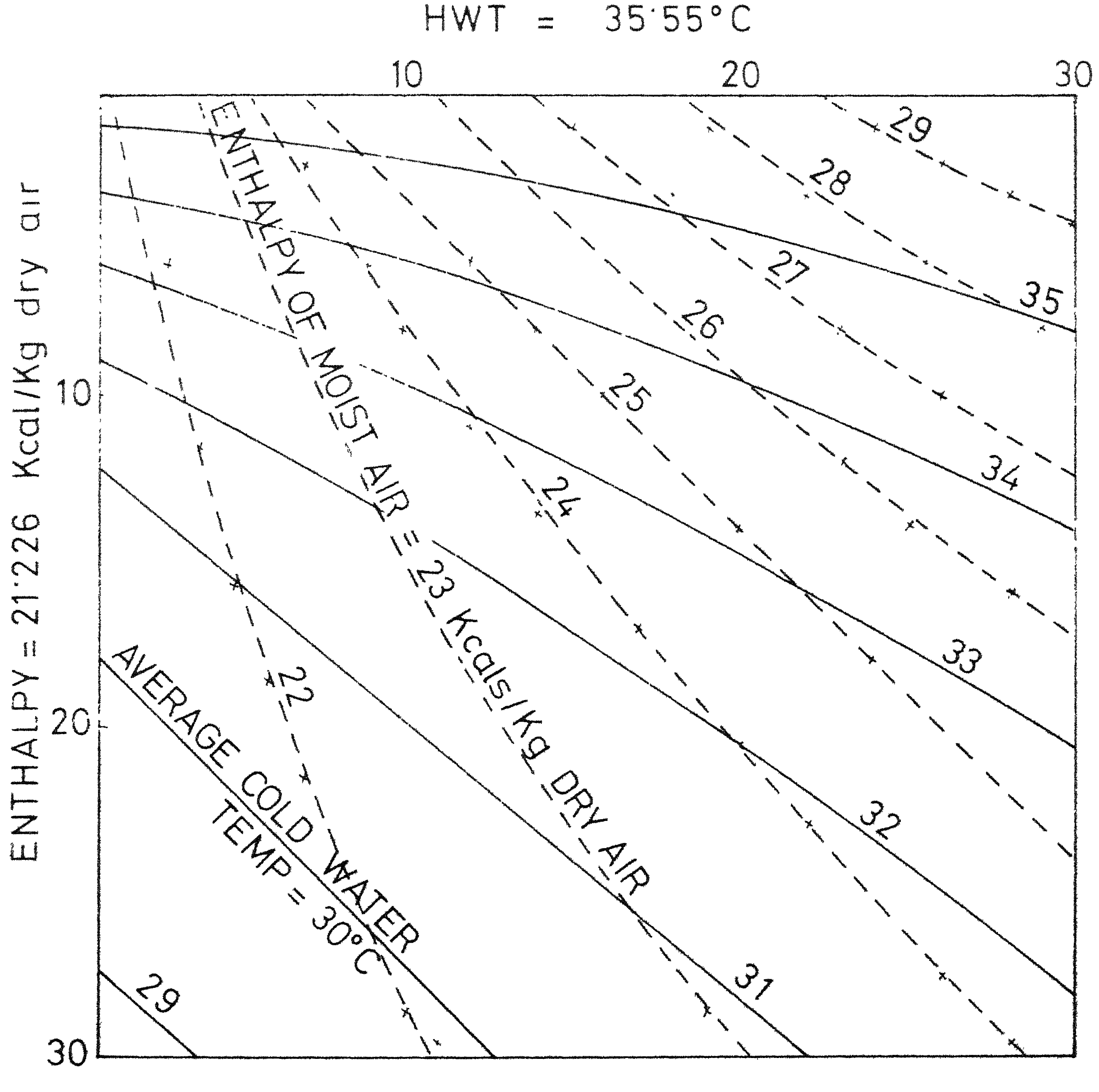


FIG. 3·26 CROSS FLOW TOWER PERFORMANCE CURVES.
(AIR CONDITIONING PLANT)

PLACES - AMRITSAR, ALLAHABAD, ASANSOL, CALCUTTA, DELHI, GAYA, GAUHATI, JAMSHEDPUR, KANPUR, LUCKNOW, MADRAS, PATNA, VISHAKAPATTANAM.

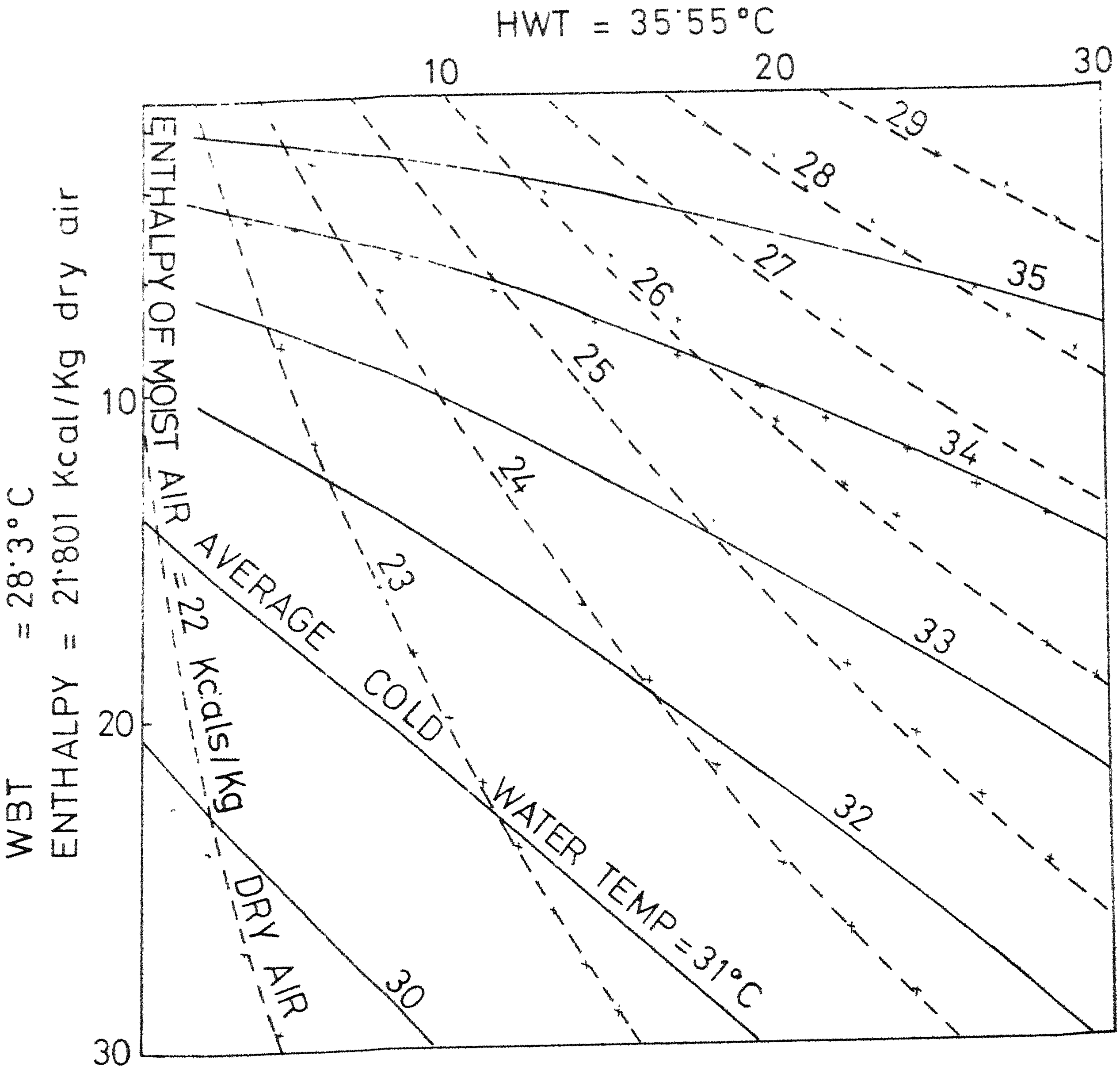


FIG. 3·27 CROSS FLOW TOWER PERFORMANCE CURVES.
(AIR CONDITIONING PLANT)

26.2°C, 27.8°C and 28.3°C to suit various industrial applications for different locations of the country.

CHAPTER IV

RESULTS AND DISCUSSION

The Tchebyshev's method for numerically, evaluating the Merkel equations (2.5) and (2.5)' is consistent over a wide variety of cooling ranges and wet bulb temperatures. As described in the previous chapter, this method has been applied to determine the cooling tower characteristic for various computation ranges. Performance curves are drawn for these ranges both for counter flow and cross flow towers.

Figures (3.2) - (3.9) show sets of curves for the counter flow tower, where the cold water temperature has been plotted as a function of the design wet bulb temperature for different cooling ranges. The design conditions are shown by dotted lines.

Four design wet bulb temperatures 24.5°C, 26.2°C, 27.8°C and 28.3°C have been considered for various locations of the country Table (3.1). From the performance curves, the cold water temperature can be obtained at these wet bulb temperatures for a particular cooling range depending on the nature of the industry. The cooling range for various industries is as following [27]:

3.5°C - 11°C	for air conditioning and refrigeration systems
5.5°C - 11°C	for diesel engine cooling
6.7°C - 11°C	for steam surface condensers

8.5°C – 16.5°C for various industrial processes.

The following examples will explain how the performance curves may be used by the manufacturer or the buyer.

Counter Flow Tower:

Example 1 : It is desired to design a counter flow cooling tower for a thermal power plant in Kanpur where the design wet bulb temperature is 28.3°C. The following data are known for the thermal power plant:

Design dry bulb temperature at Kanpur = 41.5°C.

Cooling range = 8°C.

Height of the packing in the cooling tower, V = 4 meters.

Quantity of heat to be dissipated = 100×10^6 kcal/hr.

Consider a packing of rectangular slats as shown in figure (2.2). The rectangular slats are made of wood. Choose water flow rate, L = 10,000 kg/hr.m^2 and air flow rate, G = 5500 kg/hr.m^2. Therefore, for the water to-air flow rate ratio $\frac{L}{G}$ = 1.8, the volumetric mass transfer coefficient for the packing considered is given by,

Ka = 2270 kg/m^3.hr (kg of water/kg of dry air), from table (2.1).

$$\therefore \frac{K\,a\,V}{L} = \frac{2270 \times 4}{10,000} = 0.901$$

From the counter flow tower performance curves for the thermal power plant figure (3.2), the outlet cold water

temperature at a wet bulb temperature of 28.3°C is = 35°C when $\frac{L}{G}$ ratio = 1.8 and $\frac{K\,a\,V}{L}$ = 0.901. Since, the cooling range is 8°C, the temperature of the inlet hot water is 43°C. The quantity of heat to be dissipated = 100 x 10^6 kcal/hr.

∴ Area of the cooling tower cross section for a chosen water flow rate of 10,000 kg/hr.m^2 is:

$$A = \frac{Q}{C_w \times \Delta T \times L} = \frac{100 \times 10^6}{8.0 \times 10{,}000}$$

$$= 1250\ m^2$$

Now, for the packing considered in figure 2.2), the total number of fill deck layers = $\frac{\text{height of the packing}}{\text{distance between two successive slats}}$

$$= \frac{4.0}{0.38} = 11.0$$

∴ Total air flow rate required to maintain the outlet and inlet water temperature at 35°C and 43°C, respectively throughout the tower cross section, is =

$$5500 \times 1250$$
$$= 6.87 \times 10^6\ kg/hr.$$

Density of air at 41.5°C DBT = 1.1255 kg/m^3

∴ Total volume air flow rate = $\frac{6.87 \times 10^6}{1.1255 \times 60}$ m^3/minute

$$= 1.016 \times 10^5\ m^3/minute$$

∴ Total H.P. required to drive the fans =

$$= \frac{1.016 \times 10^5}{226.536^*} \simeq 450.0\ H.P.$$

---

* A rule of thumb calculation for induced draft towers is that each 226.536 m^3/minute of air exhausted requires one horsepower [3].

The cooling tower can be divided into many cells to ensure the continuous cold water supply even if one cell breaks down. Let us assume in this case that there are four cells. Thus, four fans of about 120 H.P. each are required to be provided in the four cells so that the total H.P. = 450.00 as determined above. In order now to verify that the tower characteristic calculated in this example is the appropriate one, we use the performance curves given in figure (3.13), for the following data : WBT = 28.3°C, range = 8°C, approach = (35°C - 28.3°C = 6.7°C), and $\frac{L}{G}$ ratio = 1.8.

As it is clear from figures (3.2) - (3.9), the wet bulb temperature has an effect on the outlet cold water temperature if the cooling range is kept constant. The cold water temperature increases with the increase of the wet bulb. It was found in the above example, that for a WBT = 28.3°C, the cold water temperature is = 35°C. If the WBT's are changed to 24.5°C, 26.2°C and 27.8°C which refer to different locations in the country, the outlet cold water temperatures will be 32.8°C, 33.8°C and 34.7°C respectively, for the same cooling range.

Locations (WBT)°C	24.5	26.2	27.8	28.3
Cold water temperature leaving °C	32.8	33.8	34.7	35.0
Approach to the wet bulb, °C	8.3	7.6	6.9	6.7

In figure (3.2), the L/G ratio, $\frac{K\,a\,V}{L}$ and the packing design remains the same, for different cooling ranges. Hence, for a constant tower size, the counter flow cooling tower performs better at a wet bulb temperature of 28.3°C than at other wet bulb temperatures shown above because the approach at this wet bulb is minimum = 6.7°C. Thus, it is the characteristic of a mechanical draft tower to give better performance at higher wet bulbs.

Similarly, figures (3.3) - (3.9) and (3.10) - (3.12) may be used to determine the unknown parameters and thereby, to design counter flow cooling towers for different applications and at different locations.

It is to be noted from figures (3.10) - (3.13), that the approach to the wet bulb increases as the L/G ratio increases for a set of constant values of $\frac{K\,a\,V}{L}$, WBT and range.

Cross Flow Tower: Figures (3.16) - (3.27) show performance curves for the cross flow cooling towers. The entering hot water temperatures of 43°C, 45.2°C and 35.5°C and the inlet air wet bulb temperatures of 24.5°C, 26.2°C, 27.8°C and 28.3°C are considered. The curves indicate that for a particular industrial application, the outlet cold water temperature increases with the increase of the wet bulb temperature, if the temperature of the inlet hot water is kept constant.

These curves are dimensionless and are not related to any particular tower design. This allows the results to be applied for any cross flow tower for which K_a, G and L are known.

The following formulae are used to calculate the dimensionless co-ordinates $\bar{X}$ and $\bar{Z}$ of various positions in the tower packing in the X and Z directions, for known values of K_a, G and L.

$$\Delta \bar{X} = \frac{K_a \Delta X}{G} \qquad (3.21)$$

$$\therefore \bar{X} = \frac{K_a X}{0.06\, G} \qquad (3.21)'$$

where,

X = Packing depth, in meters in the direction of air flow.

The value 0.06 in the above equation appears as a result of the mesh size chosen for the calculation.

Similarly,

$$\Delta \bar{Z} = \frac{K_a \Delta Z}{L} \qquad (3.27)$$

$$\bar{Z} = \frac{K_a Z}{0.06\, L} \qquad (3.27)'$$

where,

Z = Packing height in meters.

In order to explain how the cross flow tower performance curves are used, we consider the same example as stated for the counter flow tower case, where the water flow rate, L = 10,000 kg/hr.m^2 and the air flow rate, G = 5500 kg/hr.m^2.

For the above water-to-air flow rate ratio $\frac{L}{G} = 1.8$,

the volumetric heat transfer coefficient, for the packing being considered, is K_a = 4500 kcal/hr.m^3 (kcal/kg of dry air) as explained in Appendix-D. Let the packing depth X, and the packing height Z be 1.5 meters and 4.0 meters, respectively.
Using relations $(3.21)'$ and $(3.27)'$,

$$\bar{X} = \frac{4500}{0.06} \times \frac{1.5}{5500}$$

$$= 20.0$$

$$\bar{Z} = \frac{4500}{0.06} \times \frac{4.0}{10000}$$

$$= 30.0$$

Therefore, in figure (3.19), the proposed cooling tower would be represented by the rectangle bounded by the lines $\bar{X}$ = 0.0, $\bar{X}$ = 20.0 and $\bar{Z}$ = 0.0, $\bar{Z}$ = 30.0. The average cold water temperature shown at the point ($\bar{X}$ = 20.0, $\bar{Z}$ = 30.0) is about 33.3°C. This means that a tower having a dimensionless height $\bar{Z}$ of 30.0 and a dimensionless depth $\bar{X}$ of 20.0 would discharge water at about 33.3°C when operating at 28.3°C wet bulb temperature and 43°C of inlet water temperature. Therefore the approach in this example is 5.0°C. The plan area of the tower is obtained as follows:
Quantity of heat to be dissipated = 100 x 10^6 kcal/hr.
Hot water temperature = 43°C.
Cold water temperature = 33.3°C, from figure (3.19).
∴ Cooling range = (43.0°C - 33.3°C)
= 9.7°C
Water flow rate, L = 10,000 kg/hr.m^2

$\therefore$ Plan area of the cross flow cooling tower $= \dfrac{Q}{C_w \times \Delta T \times L} = \dfrac{100 \times 10^6}{9.7 \times 10{,}000}$

$= 1030 \ m^2$

Hence, comparing examples 1 and 2 we find that for the same inlet hot water temperature of 43°C, for the same amount of heat dissipation and for the same packing height, the cold water temperatures for counter flow and cross flow cooling towers are 35°C and 33.3°C, respectively. Thus, a lower temperature is obtained in the cross flow cooling tower as compared to that in a counter flow installation. Also, the ground areas covered by the counter flow and cross flow cooling towers are 1250 m^2 and 1030 m^2, respectively for similar conditions. Hence, to get the same cold water temperature by the two types of cooling towers, the cross flow tower requires a smaller area. There is a reduction of nearly 25 per cent in size when compared to the counter flow cooling tower. This proves the well experienced fact that considering all the factors, the cross flow towers are economical and give better performance than the counter flow towers.

The cross flow tower performance curves are useful in predicting the unknown parameters for given design conditions and hence in designing the cross flow cooling towers.

CONCLUSION:

Performance curves have been drawn and analysed

both for the counter flow and cross flow cooling towers for the design conditions suitable for the thermal power plants, fertilizer plants and the air conditioning plants at Kanpur. Only a particular cooling range of $8^{\circ}C$ is chosen in drawing the performance curves for $(\frac{K\,a\,V}{L})$ vs: $\frac{L}{G}$ in figures (3.10) - (3.13).

More locations and cooling ranges have not been considered only to avoid the thesis becoming too voluminous. The methods described herein and the computer programs developed thereby may, however, be used to obtain performance curves for any situation. An acceptable degree of accuracy in drawing these performance curves has been maintained. Tolerance of $0.5^{\circ}C$ in approach and a tolerance of two to three per cent in the value of $\frac{K\,a\,V}{L}$ have been allowed.

The numerical methods employed and the computer programs developed to obtain the performance curves have been explained.

The present day cooling towers require careful designing and selection because of the increased costs and constraints on the availability of space. The performance curves developed in the present work should be of great use to the cooling tower manufacturers and buyers in the country in predicting the tower performance and in its selection.

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APPENDIX-A

```
 IBJOB
 IBFTC MAIN
C      ***********************************************************************
C      THIS PROCEDURE IS TO DRAW THE PERFORMANCE CURVES SHOWING COLD
C      WATER TEMPERATURE AS A FUNCTION OF WET BULB TEMPERATURE FOR VARIOUS
C      RANGES.
C      ALG=WATER TC AIR RATIO.
C      H1=ENTHALPY OF ENTERING AIR,KCAL/KG DRY AIR.
C      KAV/L=COOLING TOWER CHARECTRISTICS.
C      RA=COOLING RANGE,DEG CENTIGRADE.
C      TW=WET BULB TEMPERATURE OF THE ENTERING AIR,DEG CENTIGRADE.
C      T2=TEMPERATURE OF COLD WATER LEAVING,DEG CENTIGRADE.
C      TOL=TOLERANCE FOR KAV/L.
C      VCON=DESIGN VALUE OF KAV/L.
C      ***********************************************************************

      DIMENSION T2(50),TW(50),RA(50),H1(50)
      COMMON/CMD/R(150),A(150),B(150),NRR
 100  FORMAT(1X,131(1H*))
 102  FORMAT(1X,60(1H*),*INPUT DATA*,60(1H*))
 106  FORMAT(8X,101(1H-))
 200  FORMAT(8F10.5)
 202  FORMAT(10I3)
 204  FORMAT( /,1X,*L/G  =*,F8.4,8X,*DESIGN (KAV)/L  =*,F8.4,8X,
     1*TOLERANCE LIMIT  =*,F8.4)
 206  FORMAT( /,8X,101(1H-),//,
     18X,*WET BULB TEMP.*,3X,*ENTERING AIR ENTHALPY*,4X,*COLD WATER TEMP
     2.*,13X,*RANGE*,18X,*KAV/L*,//,8X,101(1H-))
 208  FORMAT(   10X,F10.4,10X,F10.4,13X,F10.4,13X,F10.4,13X,F10.4)
 210  FORMAT(//,10X,*RANGE OF TEMPERATURE*,15X,*A*,14X,*B*,/)
 212  FORMAT(   5X,F10.4,4X,*TO*,4X,F10.4,5X,F10.4,5X,F10.4,5X,F10.4)
 214  FORMAT(4F16.8)
      PRINT100
      PRINT 102
      READ 202,N1,N2,NR,NRR
      PRINT202,N1,N2,NR,NRR
      PRINT100
      READ 200,(H1(K),K=1,N1)
      PRINT200,(H1(K),K=1,N1)
      PRINT100
      READ 200,(TW(K),K=1,N1)
      PRINT200,(TW(K),K=1,N1)
      PRINT100
      READ 200,(T2(K),K=1,N2)
      PRINT200,(T2(K),K=1,N2)
```

```
      PRINT100
      READ 200,(RA(K),K=1,NR)
      PRINT200,(RA(K),K=1,NR)
      PRINT100
      R(1)=0.0
      NN=NRR-1
      PRINT 210
      DO 4 K=1,NN
      L=K+1
      READ214,R(L),A(K),B(K)
      PRINT 212,R(K),R(L),A(K),B(K)
    4 CONTINUE
      PRINT 100
      READ 200,ALG,VCON,TOL
      PRINT204,ALG,VCON,TOL
      PRINT 100
      PRINT206
      DO 1 I=1,N1
      DO 2 J=1,N2
      DO 3 K=1,NR
      T1=T2(J)+RA(K)
      H2=H1(I)+RA(K)*ALG
      TT=T2(J)
      RR=RA(K)
      HH=H1(I)
      CALL DELTA(T1,TT,HH,H2,ALG,RR,V)
      TEST=V-VCON
      IF(ABS(TEST).GT.TOL) GO TO 3
      PRINT208,TW(I),H1(I),T2(J),RA(K),V
    3 CONTINUE
    2 CONTINUE
    1 CONTINUE
      STOP
      END

$IBFTC DELTA
C     ************************************************************************
C     SUBROUTINE DELTA CALCULATES THE VALUE OF KAV/L,THE COOLING TOWER CHARECT
C     RISTICS BY THE TCHEBYCHEFF METHOD)
C     ************************************************************************

      SUBROUTINE DELTA(T ,TT,HH,H2,ALG,RR,V)
      SUM=0.0
      DO 1 K=1,4
      RRR=RR*ALG
      GO TO(2,3,4,5),K
    2 T =TT+0.1*RR
      HA=HH+0.1*RRR
      GO TO 6
    3 T =TT+0.4*RR
```

```
      HA =HH+0.4*RRR
      GO TO 6
    4 T =T1-0.4*RR
      HA=H2-0.4*RRR
      GO TO 6
    5 T =T1-0.1*RR
      HA=H2-0.1*RRR
    6 CALL ALPHA(T,HW)
      HD =HW-HA
    1 SUM=SUM+1./HD
      V  =RR*SUM*0.250
      RETURN
      END

 IBFTC ALPHA
C     ************************************************************************
C     THE SUBROUTINE ALPHA CALCULATES THE ENTHALPY HW FOR THE TEMP,TW        *
C     A AND B ARE CONSTANTS IN HW=A(TW)+B                                    *
C     ************************************************************************

      SUBROUTINE ALPHA(T,HW)
      COMMON/CMD/R(150),A(150),B(150),NR
      DO 1 K=1,NR
      IF(T.LE.R(K)) GO TO 2
      GO TO 1
    2 J=K-1
      GO TO 4
    1 CONTINUE
      J=0
    4 HW=A(J)*T+B(J)
      RETURN
      END
 ENTRY
```

INPUT DATA

16 36 24 91

10.04	10.71	11.42	12.16	12.92	13.72	14.55	15.41
16.32	17.25	18.23	19.26	20.33	21.45	22.62	23.83

15.0	16.0	17.0	18.0	19.0	20.0	21.0	22.0
23.0	24.0	25.0	26.0	27.0	28.0	29.0	30.0

15.0	16.0	17.0	18.0	19.0	20.0	21.0	22.0
23.0	24.0	25.0	26.0	27.0	28.0	29.0	30.0
31.0	32.0	33.0	34.0	35.0	36.0	37.0	38.0
39.0	40.0	41.0	42.0	43.0	44.0	45.0	46.0
47.0	48.0	49.0	50.0				

0.0	1.0	2.0	3.0	4.0	5.0	6.0	7.0
8.0	9.0	10.0	11.0	12.0	13.0	14.0	15.0
16.0	17.0	18.0	19.0	20.0	21.0	22.0	23.0

R(K)	A(K)	B(K)
1.00000000	0.41000000	2.25600001
2.00000000	0.42399999	2.24200001
3.00000000	0.43599999	2.21799999
4.00000000	0.45199999	2.17000008
5.00000000	0.46100003	2.13399982
6.00000000	0.48000002	2.03900003
7.00000000	0.49199998	1.96700001
8.00000000	0.50699997	1.86200047
9.00000000	0.53100002	1.66999960
10.00000000	0.54500002	1.54399967
11.00000000	0.56500000	1.34399986
12.00000000	0.58500004	1.12399960
13.00000000	0.60999990	0.82400131
14.00000000	0.62900007	0.57699871
15.00000000	0.65699995	0.18499947
16.00000000	0.67000008	-0.01000023
17.00000000	0.70999992	-0.64999962
18.00000000	0.74000001	-1.16000175
19.00000000	0.75999999	-1.51999855
20.00000000	0.80000007	-2.28000259
21.00000000	0.82999992	-2.87999725
22.00000000	0.86000001	-3.50999832
23.00000000	0.90999997	-4.61000061
24.00000000	0.93000007	-5.07000351
25.00000000	0.98000002	-6.27000046
26.00000000	1.02999997	-7.51999664
27.00000000	1.06999993	-8.55999756
28.00000000	1.12000012	-9.90999603
29.00000000	1.16999984	-11.30999756
30.00000000	1.21000004	-12.47000122
31.00000000	1.27999997	-14.56999969
32.00000000	1.34000015	-16.43000031
33.00000000	1.39999986	-18.34999847
34.00000000	1.47000003	-20.66000366
35.00000000	1.52999997	-22.69999695
36.00000000	1.61000013	-25.50000000
37.00000000	1.69000006	-28.38000488
38.00000000	1.76999998	-31.33999634
39.00000000	1.86000013	-34.76000977
40.00000000	1.93999958	-37.87998962

41.00000000	2.05000019	-42.27999878
42.00000000	2.15000010	-46.38000488
43.00000000	2.25000000	-50.58000183
44.00000000	2.36999989	-55.73997498
45.00000000	2.49000025	-61.02001953
46.00000000	2.62999964	-67.31997681
47.00000000	2.76000023	-73.29998779
48.00000000	2.90999985	-80.34997559
49.00000000	3.05999994	-87.54998779
50.00000000	3.21999979	-95.38998413
51.00000000	3.41000080	-104.89004517
52.00000000	3.58999920	-114.06994629
53.00000000	3.78000069	-123.95004272
54.00000000	4.00000000	-135.60998535
55.00000000	4.22999954	-148.02996826
56.00000000	4.48000050	-161.78002930
57.00000000	4.73999977	-176.33996582
58.00000000	5.01999950	-192.29998779
59.00000000	5.34000015	-210.86004639
60.00000000	5.60000038	-226.20001221
61.00000000	6.00000000	-250.20001221
62.00000000	6.39999962	-274.59997559
63.00000000	6.69999981	-293.19995117
64.00000000	7.30000114	-331.00000000
65.00000000	7.69999886	-356.59997559
66.00000000	8.30000114	-395.60009766
67.00000000	8.79999924	-428.59997559
68.00000000	9.39999962	-468.79992676
69.00000000	10.20000076	-523.20007324
70.00000000	10.79999924	-564.59985352
71.00000000	11.80000114	-634.60009766
72.00000000	12.59999847	-691.39990234
73.00000000	13.70000076	-770.60009766
74.00000000	14.89999962	-858.19995117
75.00000000	16.10000038	-947.00024414
76.00000000	17.59999847	-1059.49975586
77.00000000	19.20000076	-1181.10009766
78.00000000	21.10000229	-1327.40014648
79.00000000	23.19999695	-1491.19970703
80.00000000	25.60000229	-1680.80029297
81.00000000	28.59999847	-1920.79980469
82.00000000	31.90000153	-2188.09985352
83.00000000	35.70000076	-2499.70019531
84.00000000	40.29999924	-2881.49951172
85.00000000	45.70000076	-3335.10009766

APPENDIX-B

```
IBJCB
IBFTC MAIN
C     ***********************************************************************
C     THIS PROGRAM IS TO COMPUTE THE VALUES FOR KAV/L AS A FUNCTION OF     *
C     L/G FOR VARIOUS APPROACHES AND FOR A CONSTANT COOLING RANGE AND      *
C     A CONSTANT WET BULB TEMPERATURE.                                      *
C     GL=VALUE OF L/G.                                                      *
C     H1=ENTHALPY OF ENTERING AIR,KCAL/KG DRY AIR.                          *
C     KAV/L=COOLING TOWER CHARECTRISTICS.                                   *
C     RA=COOLING RANGE,DEG CENTIGRADE.                                      *
C     TW=WET BULB TEMPERATURE OF THE ENTERING AIR,DEG CENTIGRADE.           *
C     T2=TEMPERATURE OF COLD WATER LEAVING,DEG CENTIGRADE.                  *
C     ***********************************************************************

      DIMENSION T2(50),TW(50),RA(50),H1(50) ,GL(50)
      COMMON/CMD/R(150),A(150),B(150),NRR
 100  FORMAT(1X,131(1H*))
 102  FORMAT(1X,60(1H*),*INPUT DATA*,60(1H*))
 200  FORMAT(8F10.5)
 202  FORMAT(10I 3)
 208  FORMAT(2X,8F16.3)
 210  FORMAT(//,10X,*RANGE OF TEMPERATURE*,15X,*A*,14X,*B*,/)
 212  FORMAT(    5X,F10.4,4X,*TO*,4X,F10.4,5X,F10.4,5X,F10.4,5X,F10.4)
 214  FORMAT(4F16.8)
 600  FORMAT(40X,92(1H-))
      PRINT100
      PRINT 102
      READ 202,N1,N2,NR,NRR,NLG
      PRINT202,N1,N2,NR,NRR,NLG
      PRINT100
      READ 200,(H1(K),K=1,N1)
      PRINT200,(H1(K),K=1,N1)
      PRINT100
      READ 200,(TW(K),K=1,N1)
      PRINT200,(TW(K),K=1,N1)
      PRINT100
      READ 200,(T2(K),K=1,N2)
      PRINT200,(T2(K),K=1,N2)
      PRINT100
      READ 200,(RA(K),K=1,NR)
      PRINT200,(RA(K),K=1,NR)
      PRINT100
      READ 200,(GL(K),K=1,NLG)
      PRINT200,(GL(K),K=1,NLG)
```

```
      R(1)=0.0
      NN=NRR-1
      PRINT100
      PRINT 110
      DO 4 K=1,NN
      L=K+1
      READ214,R(L),A(K),B(K)
      PRINT 212,R(K),R(L),A(K),B(K)
    4 CONTINUE
      PRINT 100
      DO 1 I=1,N1
      DO 2 J=1,NR
      DO 3 K=1,N2
      DO 5 L=1,NLG
      T1=T2(K)+RA(J)
      H2=H1(I)+RA(J)*GL(L)
      TT=T2(K)
      RR=RA(J)
      HH=H1(I)
      GG=GL(L)
      CALL DELTA(T1,TT,HH,H2,GG,RR,V)
      ASPRO=TT-TW(I)
      PRINT 208,H1(I),TW(I),TT,ASPRO,RA(J),GL(L),V
    5 CONTINUE
    3 CONTINUE
      PRINT 600
    2 CONTINUE
    1 CONTINUE
      STOP
      END

 IBFTC DELTA

C     ***************************************************************************
C     SUBROUTINE DELTA CALCULATES THE VALUE OF KAV/L,THE COOLING TOWER CHARECTE-
C     RISTICS BY THE TCHEBYCHEFF METHOD)                                        *
C     ***************************************************************************

      SUBROUTINE DELTA(T1,TT,HH,H2,ALG,RR,V)
      SUM=0.0
      DO 1 K=1,4
      RRR=RR*ALG
      GO TO(2,3,4,5),K
    2 T =TT+0.1*RR
      HA=HH+0.1*RRR
      GO TO 6
    3 T =TT+0.4*RR
      HA=HH+0.4*RRR
      GO TO 6
```

```
  4  T =T1-0.4*RR
     HA=H2-0.4*RRR
     GO TO 6
  5  T =T1-0.1*RR
     HA=H2-0.1*RRR
  6  CALL ALPHA(T,HW)
     HD =HW-HA
     SUM=SUM+1./HD
     V  =RR*SUM*0.25)
     RETURN
     END

 IBFTC ALPHA

C    ********************************************************************
C    THE SUBROUTINE ALPHA CALCULATES THE ENTHALPY HW FOR THE TEMP,TW     *
C    A AND B ARE CONSTANTS IN HW=A(TW)+B                                 *
C    ********************************************************************

     SUBROUTINE ALPHA(T,HW)
     COMMON/CMD/R(150),A(150),B(150),NR
     DO 1 K=1,NR
     IF(T.LE.R(K)) GO TO 2
     GO TO 1
  2  J=K-1
     GO TO 4
  1  CONTINUE
     J=0
  4  HW=A(J)*T+B(J)
     RETURN
     END
 ENTRY
```

INPUT DATA

16 36 21 91 16

10.04	10.71	11.42	12.16	12.92	13.72	14.55	15.41
16.32	17.25	18.23	19.26	20.33	21.45	22.62	23.83
15.0	16.0	17.0	18.0	19.0	20.0	21.0	22.0
23.0	24.0	25.0	26.0	27.0	28.0	29.0	30.0
15.0	16.0	17.0	18.0	19.0	20.0	21.0	22.0
23.0	24.0	25.0	26.0	27.0	28.0	29.0	30.0
31.0	32.0	33.0	34.0	35.0	36.0	37.0	38.0
39.0	40.0	41.0	42.0	43.0	44.0	45.0	46.0

47.0	48.0	49.0	50.0				

0.0	1.0	2.0	3.0	4.0	5.0	6.0	7.0
8.0	9.0	10.0	11.0	12.0	13.0	14.0	15.0
16.0	17.0	18.0	19.0	20.0			

0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6
1.8	2.0	2.2	2.4	2.6	2.8	3.0	3.2

FOR TEMPERATURE RANGES AND CONSTANTS A AND B, SEE APPENDIX-A *

SSSSSS

APPENDIX-C

```
IBJOB
IBFTC MAIN
C      ***********************************************************************
C      THIS PROCEDURE IS TO CALCULATE THE DEMAND CURVES FOR VARIOUS CROSS-FLOW  *
C      CONDITIONS. THE CROSS-FLOW COOLING TOWER INVOLVES A TWO-DIMENSIONAL      *
C      FLOW PATTERN IN WHICH WATER FALLS DOWNWARD THROUGH THE TOWER AND AIR IS  *
C      DRAWN HORIZONTALLY THROUGH THE PACKING.                                  *
C      HA=ENTHALPY OF SATURATED AIR,KCAL/KG DRY AIR.                            *
C      HE=ESTIMATED ENTHALPY,KCAL/KG.                                           *
C      HX,HAX=MESH SIZE,DIMENSIONLESS,A VALUE OF 0.06 IS SELECTED.              *
C      HW=ENTHALPY OF SATURATED AIR AT WATER TEMPERATURE,KCAL/KG DRY AIR.       *
C      HZ,HAZ=MESH SIZE,DIMENSIONLESS,A VALUE OF 0.06 IS SELECTED.              *
C      TW=TEMPERATURE OF WATER,DEG CENTIGRADE.                                  *
C      TWB=WET BULB TEMPERATURE OF THE ENTERING AIR.(DEG CENTIGRADE)            *
C      TWE=ESTIMATED TEMPERATURE,DEG CENTIGRADE)                                *
C      ***********************************************************************

       DIMENSION HA(30,30),HW(30,30),TW(30,30),TWE(30,30),HE(30,30),
      1HD(30,30),W(30,30)
       COMMON/CMD/R(150),A(150),B(150)
 48    FORMAT(5X,*USES THE SECOND APPROXIMATION*)
 200   FORMAT(1X,131(1H*))
 201   FORMAT(1X,60(1H*),*INPUT DATA*,60(1H*))
 210   FORMAT(//,10X,*RANGE OF TEMPERATURE*,15X,*A*,14X,*B*,/)
 212   FORMAT(   5X,F10.4,4X,*TO*,4X,F10.4,5X,F10.4,5X,F10.4,5X,F10.4)
 214   FORMAT(4F16.8)
 215   FORMAT(3F7.3)
       PRINT200
       PRINT201
       NRR=91
       R(1)=0.0
       NN=NRR-1
       PRINT 210
       DO 4 K=1,NN
       L=K+1
       READ214,R(L),A(K),B(K)
       PRINT 212,R(K),R(L),A(K),B(K)
   4   CONTINUE
       PRINT200
       READ 215,HA(1,1),HW(1,1),TW(1,1)
       PRINT 215,HA(1,1),HW(1,1),TW(1,1)
       PRINT200
       DO 30 I= ,30
 30    HW(I,1)=HW(1,1)
       DO 31 J= ,30
```

```
31    HA(1,J)=HA(1,1)
      DO 13 IE=2,30
13    TW(IE,1) =TW(1,1)
      DO 10 IX=2,30
      IY=IX-1
      HX=0.0
10    HA(IX,1)=HA(IY,1)+(HX/2.)*(HW(IY,1)+HW(IX,1)-HA(IY,1))/(1.+HX/2.)
      READ 2 5,TWE
      PRINT 20
      DO 14 IF=2,30
      IG=IF-
      TWE(1,IF)=(TW(1,IG)-(TW(1,IG)-TWB))/2.
      CALL ALPHA(TWE(1,IF),HW(1,IF))
      HZ=0.06
14    TW(1,IF)=TW(1,IG)-(HZ/2.)*(HW(1,IG)+HW(1,IF)-HA(1,IG)-HA(1,IF))
18    DO 15 IH=2,30
      IF(ABS(TW(1,IH)-TWE(1,IH)).LE.0.01) GO TO 15
      PRINT 48
      GO TO 16
15    CONTINUE
      GO TO 19
16    DO 17 IJ=2,30
      IA=IJ-1
      TWE(1,IJ)=(TW(1,IJ)+TWE(1,IJ))/2.
      CALL ALPHA(TWE(1,IJ),HW(1,IJ))
17    TW(1,IJ)=TW(1,IA)-(HZ/2.)*(HW(1,IA)+HW(1,IJ)-HA(1,IA)-HA(1,IJ))
      GO TO 18
C     ************************************************************************
C   TO CALCULATE THE VALUE AT THE INTERIOR POINTS KNOWING VALUES OF TW,HA,HW   *
C     FROM PRECEDING POINTS                                                    *
C     ************************************************************************

19    DO 25 IT=2,30
      IU=IT-1
      DO 25 IV=2,30
      II=IV-1
      HE(IT,IV)=HW(IU,IV)-HA(IU,IV)
      HAX=0.06
      HAZ=0.06
      HA(IT,IV)=HA(IU,IV)+(HAX/2.)*(HW(IU,IV)-HA(IU,IV)+HE(IT,IV))
      TW(IT,IV)=TW(IT,II)-(HAZ/2.)*(HW(IT,II)-HA(IT,II)+HE(IT,IV))
      CALL ALPHA(TW(IT,IV),HW(IT,IV))
25    HD(IT,IV)=HW(IT,IV)-HA(IT,IV)
40    DO 21 IP=2,30
      DO 21 IQ=2,30
      IF(ABS(HD(IP,IQ)-HE(IP,IQ)).GT.0.01) GO TO 22
21    CONTINUE
      GO TO 23
22    DO 24 IK=2,30
```

```
      IM=IK-1
      DO 24 IL =2,30
      IN=IL-
      HE(IK,IL)=(HE(IK,IL)+HD(IK,IL))/2.
      HA(IK,IL)=HA(IM,IL)+(HAX/2.)*(HW(IM,IL)-HA(IM,IL)+HE(IK,IL))
      TW(IK,IL)=TW(IK,IN)-(HAZ/2.)*(HW(IK,IN)-HA(IK,IN)+HE(IK,IL))
      CALL ALPHA(TW(IK,IL),HW(IK,IL))
 24   HD(IK,IL)=HW(IK,IL)-HA(IK,IL)
      GO TO
 23   CALL SIGMA(TW)
      PRINT2
      CALL SIGMA(HA)
      PRINT2
  6   CONTINUE
      STOP
      END

 IBFTC ALPHA

C     ************************************************************************
C     THE SUBROUTINE ALPHA CALCULATES THE ENTHALPY HW FOR THE TEMP,TW        *
C     A AND B ARE CONSTANTS IN HW=A(TW)+B                                    *
C     ************************************************************************

      SUBROUTINE ALPHA(X,Y)
      COMMON/CMD/R(150),A(150),B(150)
      NR=91
      DO 1 K=1,NR
      IF(X.LE.R(K))GO TO 2
      GO TO 1
  2   J=K-1
      GO TO 4
  1   CONTINUE
      J=C
  4   Y=A(J)*X+B(J)
      RETURN
      END

 IBFTC SIGMA

C     ************************************************************************
C     THE SUBROUTINE SIGMA GROUPS THE REQUIRED ELEMENTS IN A 30*30 MATRIX    *
C     ACCORDING TC TEMPERATURE TW AND ENTHALPY HA)                           *
C     ************************************************************************

      SUBROUTINE SIGMA(TW)
      DIMENSION TW(30,30)
      TWMA=TW(1,1)
      TWMI=TW(1,1)
      DO 1 K=1,30
```

```
      DO 2 L=1,30
      IF(TW(K,L).GT.TWMA) TWMA=TW(K,L)
      IF(TW(K,L).LT.TWMI) TWMI=TW(K,L)
    2 CONTINUE
    1 CONTINUE
      N1=IFIX(TWMA)
      N2=IFIX(TWMI)
      N=N1-N2+1
      A=FLOAT(N1)+1.
      DO 225 II=1,N
      A=A-1.
      DO 227 KK=1,30
      DO 226 LL=1,30
      B=TW(KK,LL)
      EPSI=0.10
      IF(ABS(A-B).GT.EPSI) GO TO 226
      PRINT 228, KK,LL,B
  226 CONTINUE
  227 CONTINUE
      PRINT 400
  228 FORMAT(5X,*TW(*,I2,*,*,I2,*)=*,2X,F6.2)
  400 FORMAT(1X,30(1H-))
  225 CONTINUE
      RETURN
      END
 ENTRY
```

INPUT DATA

```
      ENTERING AIR ENTHALPY,DEG CENTRIGRADE
      ENTERING WATER TEMPERATURE, DEG CENTIGRADE
      WET BULB TEMPERATURE OF ENTERING AIR,DEG CENTIGRADE
----------------------------------------------------------------------------
C     FOR TEMPERATURE RANGES AND CONSTANTS A AND B, SEE APPENDIX-A          *
----------------------------------------------------------------------------

SSSSSS
```

APPENDIX - C_2

Enthalpy of air water vapour mixture at saturation [28]
(0^oC Datum)

Temp. (^oC)	Enthalpy(h) (kcal/kg)	Temp. (^oC)	Enthalpy(h) (kcal/kg)	Temp. (^oC)	Enthalpy(h) (kcal/kg)
0	2.256	30	23.83	60	109.8
1	2.[illegible]6	31	25.11	61	115.8
2	3.090	32	26.45	62	122.2
3	3.526	33	27.85	63	128.9
4	3.978	34	29.32	64	136.2
5	4.439	35	30.85	65	143.9
6	4.919	36	32.46	66	152.2
7	5.411	37	34.15	67	161.0
8	5.918	38	35.92	68	170.4
9	6.449	39	37.78	69	180.6
10	6.994	40	39.72	70	191.4
11	7.559	41	41.77	71	203.2
12	8.144	42	43.92	72	215.8
13	8.754	43	46.17	73	229.5
14	9.383	44	48.54	74	244.4
15	10.04	45	51.03	75	260.5
16	10.71	46	53.66	76	278.1
17	11.42	47	56.42	77	297.3
18	12.16	48	59.33	78	318.4
19	12.92	49	62.39	79	341.6
20	13.72	50	65.61	80	367.2
21	14.55	51	69.02	81	395.8
22	15.41	52	72.61	82	427.7
23	16.32	53	76.39	83	463.4
24	17.25	54	80.39	84	503.7
25	18.23	55	84.62	85	549.4
26	19.26	56	89.10	86	602.3
27	20.33	57	93.84	87	663.4
28	21.45	58	98.86	88	734.7
29	22.62	59	104.2	89	819.1

APPENDIX-D

Volumetric heat transfer coefficient [16]: When it is very difficult to determine accurately the free surface of the liquid, e.g., on breaking up the flow of circulating water into droplets, the volumetric of heat and mass-transfer coefficients are used, i.e., coefficients that are based, not on the unit surface of the water, but on the unit active volume of the cooling tower. The volumetric heat transfer coeffic-

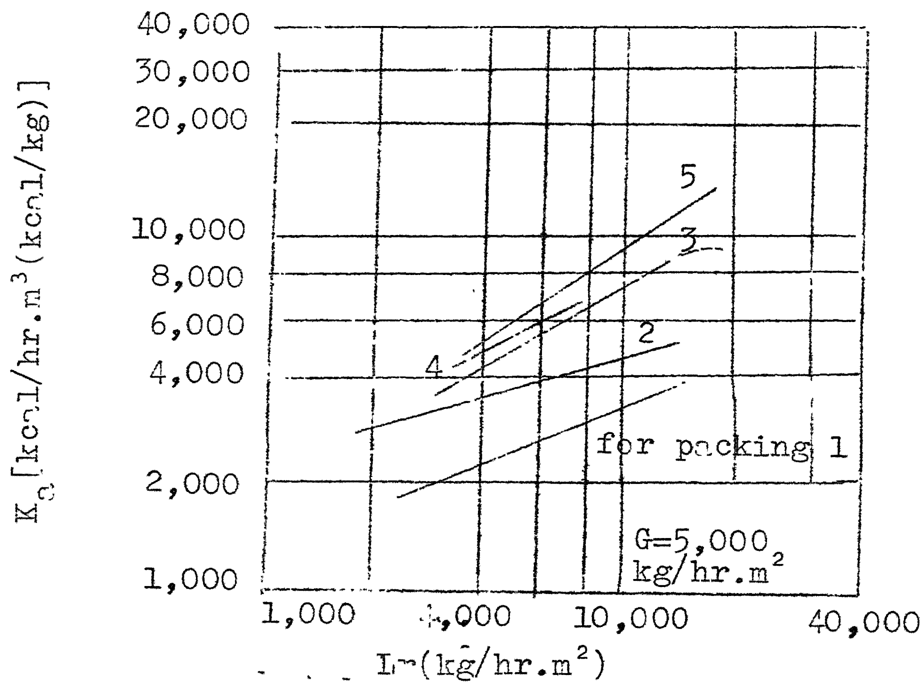

Fig.(D-1), Comparison of performance characteristics of various packings (details of packing are given in table D-1).

ient K_a is dependent upon so many factors like air flow rate, water flow rate, type of packing etc.

??????

Various investigators have obtained the value for K_a experimentally and they have been presented in figure (D-1). The details of packings used by these investigators are given in table (D-1).

Table (D-1)

Sl. No.	Investigator	Packing	Horizontal Spacing	Vertical Spacing
1	Lowe and Christie	Corrugated asbestos, cement louvres	53 mm	143 mm
2	Lichtenstein	Wooden slats (10x8) mm	10 mm	380 mm
3	Uchida et al	Honey comb board, plastication paper	112 mm	65 mm
4	Simpson et al	Masonite sheet	15 mm	-
5	Uchida and Sharah	Inclined wooden slats (10 mm)	72 mm	67 mm